SUBMERSIBLE ORIENTED PLATFORM FOR DEEP OCEAN SEDIMENT STUDIES (SOPDOSS)

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THESIS

SUBMERSIBLE ORIENTED PLATFORM
FOR DEEP OCEAN SEDIMENT STUDIES
(SOPDOSS)

by

Maurice LeRoy Hooks

June 1974

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(20. ABSTRACT continued)

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Submersible Oriented Platform
for Deep Ocean Sediment Studies
(SOPDOSS)

by

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Lieutenant, United States Navy
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This thesis presents the design, fabrication, and operation of a deep ocean sediment testing platform which is operated by a submersible to depths of 8000 feet. The platform has a variable ballast system and is transportable along the ocean floor by the submersible. It has an in-air weight of approximately 15,000 pounds and is capable of maintaining neutral buoyancy. The platform is capable of taking and storing eight individual cores and is equipped with eight instrument drives, each capable of inserting a 3 inch diameter probe to a depth of 3 feet in any normal ocean sediment for in situ measurements. The platform is lowered to the ocean floor by cable and is capable of independent surfacing. The platform has a 9 foot square base and is approximately 18 feet high including the main float. In high shear strength sediments, the platform is capable of providing 5000 pounds of hold down force at a depth of 8000 feet.

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DEDICATION AND RECOGNITIONS

The following paper is dedicated to my parents for their tolerance of my youth and to Mary Ellen for her understanding.

. . .

A special thanks to Professor Roy Prowell for his assistance in the design of the platform and support as my adviser. Thanks to Professor Robert Andrews for the incentive to begin the task of designing the platform. Thanks to the persons of SUBDEVGRP 1 and the crews of the MAXINE D and SEA CLIFF for furthering my knowledge of deep ocean operations and providing constructive criticism of the design.

I. INTRODUCTION

Recent changes in the scientific community have directed interest to the ocean's bottom rather than the moon's behind. This effort has brought about many techniques for determining the physical properties of sediments taken from the ocean's floor. The use of instruments, such as the vane shear, cone penetrometer, and sensors to measure acoustic and other physical properties of sediments, have been limited for the most part to shallow waters. The deep ocean studies have been sparse and in most cases accomplished by making sediment property determinations from cores brought to the surface. Due to the disturbance of the sediment during coring and handling, the resulting measurements are not necessarily those that would represent the "in situ" sediment. Since use of the ocean floor often requires an exacting knowledge of properties, such as dynamic shear strength (now known to be complex and frequency dependent), compressive strength, and density, as they exist in situ, efforts are being made to place the instrumentation on the ocean floor. In this way, actual phenomenon, such as acoustic propagation, can be observed with a minimal disturbance to the natural environment.

Since the process of instrumentation of the ocean's sediments is an expensive and complex endeavor, it is only reasonable that one would desire to maximize the amount of

data collected. A vehicle for such an operation would be required to carry many different instruments and be capable of taking any data that may be required. It should also be controllable in order to maximize the accuracy and dependability of its data. As with any experimental analysis, the insight and interaction of the experimenter with the experiment is essential for success; hence the requirement for a submersible.

The objective of the Submersible Oriented Platform for Deep Ocean Sediment Studies (SOPDOSS) is to provide the vehicle for in situ measurement of deep ocean sediments. The vehicle is capable of carrying several instruments to the ocean floor and providing mechanical and electrical power for their operation. It will take cores of sediment and store them for retrieval at the surface. The platform is controlled from a submersible and can be transported by the submersible for short distances along the bottom in order to take data at multiple sites. The platform has a variable ballasting capability to provide neutral buoyancy maneuverability, lowering weight, and surfacing buoyancy. It is versatile and can be oriented to nearly any desired ocean study.

SOPDOSS will provide the link to deep ocean sediment testing necessary to generate accurate and dependable information on the world's ocean sediments.

II. GENERAL BACKGROUND

A literature search revealed two noteworthy platforms, currently in use, capable of testing in situ sediments. These are the Deep Ocean Sediment Probe (DOSP), developed by the Naval Underwater Sound Lab [1]*, and the Deep Ocean Test-in-Place and Observation System (DOTIPOS) developed by the Naval Civil Engineering Lab [2]. Both of these devices are controlled from the surface and are limited in their use by the types of instrumentation they are capable of carrying.

In 1968 Westinghouse Corporation completed a feasibility study of a sediment pod (SEDOPOD) for use with a submersible [3]. The objective of the study was to provide the Naval Oceanographic Office, Deep Vehicle Branch, with information about the feasibility of constructing a platform capable of testing deep ocean sediments. A study of this report makes an interesting contrast with the SOPDOSS.

A. DEEP OCEAN SEDIMENT PROBE (DOSP)

DOSP is designed to be operated by either a submersible or a surface vessel to a depth of 5,000 ft. To date, the only published use is with a surface vessel. The platform is approximately 4 1/2 ft square at the base, approximately

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*Bracketed numbers refer to the List of References.

5 ft high, weighs 350 lb in water and 550 lb in air. Four retractable instrument probes are located at the corners of the base and a single core tube is located at the center. The probes and core are lowered into the sediment by screw drives operated by small dc motors through chain drives. Power is provided by the associated vessel.

The hold down system of the platform is a circular plate in the center of the frame. The plate rests on the sediment and has the suction of a small vane pump applied at its center. During operation, the pump provides a pressure differential across the plate and thus a force to resist the loads of probe insertion and coring. This is a significant development in deep ocean platforms and is directly incorporated in SOPDOSS.

The probes on DOSP consist of a spark source and three hydrophones for the measurement of the compression wave speed. The instrumentation of the platform is not necessarily limited to the measurement of the compression wave speed, since other instruments could be adapted to the probe assemblies. This could be expensive and the total number of drives available could not be increased without major design changes in the platform. The single core system requires that the platform be raised to the surface after each sampling. If the platform is used in deep water, this could mean as much as three hours for a single operation [1].

The platform is useful for shallow water measurements when only a small amount of data is required over a large area of ocean bottom.

B. DEEP OCEAN TEST-IN-PLACE AND OBSERVATION SYSTEM (DOTIPOS)

This platform is designed to be operated solely by a surface vessel. The platform is pyramidal, 17 1/2 ft high, with a square base, 18 ft wide. The platform weighs 2,000 lb in water and 6,200 lb in air. The basic platform has installed television cameras and lights, an electronic pressure sphere, and a transformer box. The system is designed to be operated by telemetry and powered by 120 Vac power from the surface. This requires extensive electronics both on the platform and surface vessel. The instrumentation of the platform is separate and not included in the previously stated weights. A massive combination vane shear and cone penetrometer device and a single tube coring device have been fitted to the platform. Successful tests have been performed to 5,600 ft. As with DOSP, if individual cores are desired, the platform must be raised to the surface for each operation. Other instrumentation could be adapted to the platform if a compatible instrument insertion device were constructed. The platform has ample space for such additions. The hold down force is provided by negative buoyancy alone. The main frame is fabricated from 6 in. diameter pipe of 6061-T6 aluminum [2].

The complex electronics and excessive weight of the platform, in addition to the limited scope of instrumentation, make this platform not suitable for use in extensive testing of deep ocean sediments.

C. FEASIBILITY STUDY OF SEDOPOD

The object of the study was a triangular platform approximately 5 ft on a side and 9 ft high. The platform consists of a frame of titanium, a pressure vessel of titanium, and instrumentation for sea water properties and sediment shear strength by vane shear. The platform weighs 446 lb in air and 12 lb in water, less all instrumentation. The design depth of this platform is 14,000 ft. Discussion of the design criteria and alternatives is extensive and was incorporated into the basic philosophy developing the SOPDOSS.

The study discusses three basic hold down devices; the spade, screw, and vibratory anchors. The spade anchor consists of an anchor fluke-like structure that would be forced into the sediment. The screw anchor is a simple auger which would be rotated into the sediment. The vibratory anchor consists of a pointed cylinder which is driven into the sediment, the action being similar to driving a piling. The study indicates that a vibratory motor would have to be developed for the latter. The concurrent development of DOSP perhaps provides the reason that the differential pressure plate was not considered as a hold down device.

The platform is designed to be escorted to the ocean floor by a submersible and, after operations, escorted back to the surface. While the platform is in operation, the submersible will stand off and control the instruments and make measurements through an umbilical cable. This allows for maneuverability of the submersible for observation of the tests and reduces the interference of a vehicle in close proximity. The dc power for the platform is provided by the submersible.

The platform is designed to be nearly neutrally buoyant, but since there is no variable buoyancy, the variability or addition of instrumentation is severely limited. Any variation of buoyancy would have to be compensated for by the submersible. To provide a coring capability would require a complete redesign of the platform. The estimated cost of the platform, including sensors, is $500,000.00 [3].

Although the design incorporates many of the basic requirements of a deep ocean sediment testing platform, its limited instrumentation, lack of a coring capability, and excessive cost will prohibit its use. Aluminum costs approximately 1/10 that of titanium and could easily have been used in the fabrication of the frame.

D. SUBMERSIBLES CURRENTLY AVAILABLE

Table A lists various submersibles capable of handling a platform for deep ocean sediment testing. The only submersible capable of handling a platform below 8,000 ft is <u>TRIESTE II</u>. The availability of submersibles for operation

is an important aspect of the design of the platform since the ultimate goal is the extended mapping of broad areas of the ocean floor. It would be inconsistent to demand the sole use of a single submersible for the entire study.

Table A

Available Submersibles [4]

Name	Owner	Displacement, lb	Max. Oper. Depth, ft
ALUMINANT	Reynold Sub Services, Miami	146,000	8,000
ALVIN	Woods Hole Ocean, Inst.	32,000	6,000
DEEP QUEST	Lockheed Corp.	110,000	8,000
SEA CLIFF	US Navy SubDevGr 1	48,000	6,500
TRIESTE II	US Navy SubDevGr 1	100,000	20,000
TURTLE	US Navy SubDevGr 1	48,000	6,500

The support ships for these submersibles vary considerably. The vessels supporting ALVIN, SEA CLIFF, and TURTLE have cranes which lift the submersible to the deck after operations and are certainly capable of handling a platform. The MAXINE D, which supports SEA CLIFF and TURTLE, has a 50 ton crane and a 10 ton auxiliary boom. The ALUMINANT and TRIESTE II are generally towed to the operational site and the actual vessel in support can be varied to meet the demands of the operation. DEEP QUEST has a unique catamaran hulled support

vessel which is capable of transporting the submersible in the bay between the hulls. The auxiliary equipment of the vessel is not known, but it should be capable of handling a platform.

III. DESIGN PROBLEM

The platform designed will transport sediment and deep ocean instrumentation to the ocean floor and provide electrical and mechanical power and mobility, with the assistance of a submersible, necessary to complete required testing at various sites on the ocean floor to a depth of 8,000 ft.

The instrument drive mechanisms will provide mechanical power for instruments which require insertion into the sediment or any form of vertical movement, such as placement of geophone arrays. The drives will be powered by dc motors from onboard batteries and controlled from the submersible. The drives will be easily located at many locations on the outside of the square main frame and will be easily repositioned to meet varying requirements.

The platform will be capable of taking and storing eight cores. The cores will be taken from as near the center of the platform as possible in order to make a given core as representative as possible of the testing accomplished at the platform sides.

The platform will be capable of maintaining buoyancy of ± 400 lb while submerged. The buoyancy will be controllable from the submersible. By making the platform neutrally buoyant, the submersible will be able to maneuver the platform from site to site.

All operations, both mechanical and electrical, will be controlled from the submersible through an umbilical cable. The cable will be connected upon mating of the submersible and platform on the ocean floor. The cable will be detached prior to the independent surfacing of each vehicle.

Two soft (internal pressure equals sea pressure) boxes provide for battery stowage and electric controls. Space will be provided for a hard sphere housing for components not capable of withstanding sea pressure. Basically all electrical components will be housed in soft boxes to eliminate the costly production of hard vessels.

The platform can be disassembled into reasonable pieces for transportation on land and subsequently reassembled with hand tools.

Since sea water is a hostile environment and subjects metals to serious corrosion, there are no two dissimilar metals which are in contact and subject to sea water simultaneously. With adequate washing of the platform after each operation, any appreciable corrosion should be eliminated.

The basic underlying criterion is to provide a design such that it can be fabricated in an average machine shop with a minimum of mill work, at reasonable costs in both material and labor.

IV. DESIGN CONCEPTS AND SELECTION

SOPDOSS is divided into ten fabrication groups which are the main frame, instrument drives, coring device, ballast tank, motor boxes and drive systems, battery box, control box and platform circuits, air system, hold down system, and flotation and main float. Each group is discussed in detail in this section. The basic components are labeled on Plates 1 and 2. A study of these plates will provide continuity for the following discussions.

The actual calculations and design methods are contained in detail in Appendix A.

A. MAIN FRAME

The main frame (Fig. II) is the sole supporting device for all of the platform's components such as the instrument drive, coring device, hold down plates, etc. The overall dimensions of the frame depend directly on the size of the components and their loads. After consideration of components, a base dimension of 9 ft square was chosen. This provides adequate space for all components and allows some room for additions. Since the instrument drives are located on the outside of the main frame, the frame should be large enough to provide for any linear test arrays, such as compression wave speed testing. The usual distance between the transducer and the two hydrophones is 1 meter. A 9 ft side provides adequate room.

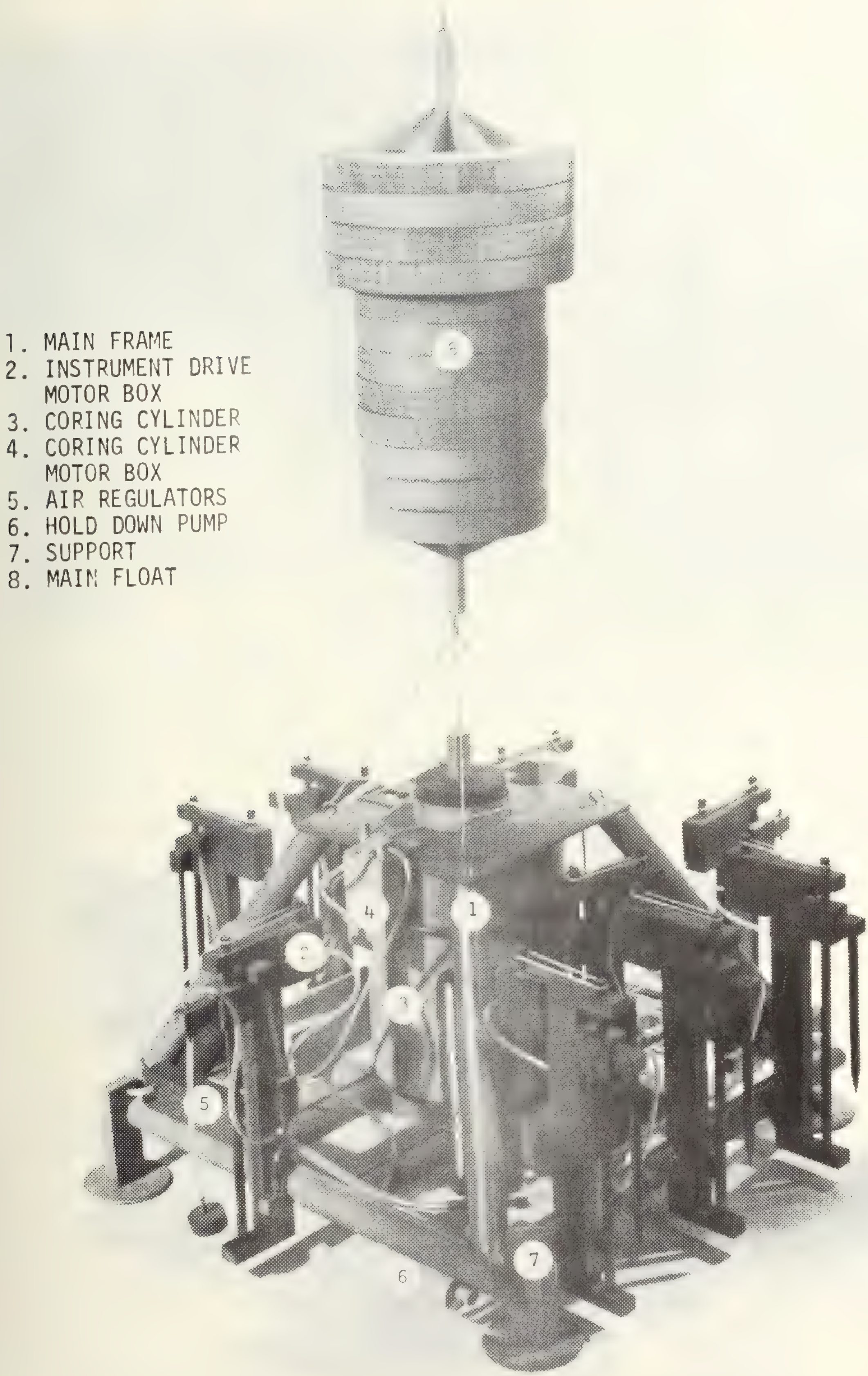


PLATE 1. MOCK-UP OF SOTLOSS, VIEW A

1. MAIN FRAME
2. INSTRUMENT DRIVE
3. BATTERY BOX
4. CONTROL BOX
5. BALLAST TANK
6. HOLD DOWN PLATE
7. UMBILICAL CONNECTOR
8. AIR FLASKS
9. MAIN FLOAT

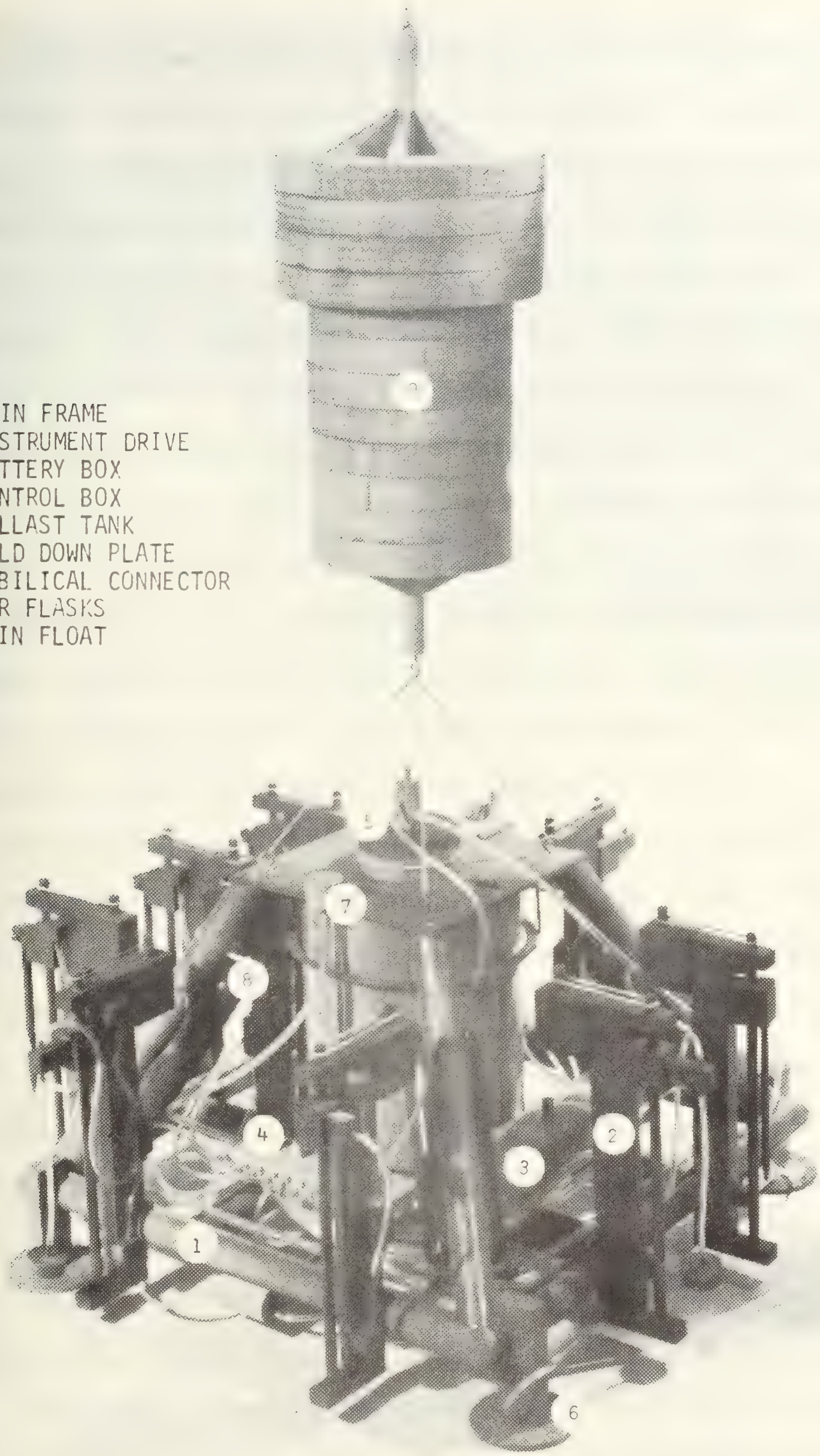


PLATE 2. MOCK-UP OF [illegible], VIEW B

The most significant loading of the main frame is due to torsion caused by eccentric loading during instrument insertion or withdrawal. The instrument drive is fixed to the main frame and any axial rotation of the frame member to which the instrument is attached will be represented as a translation of the instrument tip. In addition to causing adverse loading of the instrument, it can severely disturb the sediment being tested and cause erroneous data to be taken. An 8 in. diameter pipe, 1/4 in. wall thickness, of 6061-T6 aluminum, combines proper resistance to torque with adequate bending strength. Aluminum, 6061-T6, was chosen due to its high strength, low density, availability in structural shapes, and reasonable cost. With proper maintenance, 6061-T6 aluminum will provide good resistance to pitting. A long rectangular (cross-section 1-1/2 in. x 1 in.) bar is welded to the top and bottom of each pipe in the lower frame to facilitate fixing of the components to the frame and to add to the bending strength of the members (Fig. II-3-1).

The upright members are also of 8 in. aluminum pipe. These support the platform during lifting and provide structural stability for the coring device and ballast tank. Pipe was chosen in order to minimize the variation in structural components and to maximize the flotation volume available (Fig. II-1-2, II-2).

The main frame is sectioned by bolted flange plates (Fig. II-1-1) in order to facilitate disassembly and transportation of the platform. K-Monel bolts are used because of their high strength and corrosion resistance.

All voids in the pipes are filled with syntactic foam, a flotation material of small hollow glass spheres in an epoxy matrix. Since the pipe would have to be flooded to prevent collapse, it is consistent with the goal of neutral buoyancy that every available place should be used for flotation. The main frame, without any components, would weigh 1,939 lb in air and -182 lb in sea water.

B. INSTRUMENT DRIVE

The primary function of the instrument drive is to provide the mechanical power necessary to insert instruments into the sediment and, upon completion of testing, withdraw them. The motion required is linear, acting on a line perpendicular to the surface of the sediment.

Three basic mechanisms were considered. They were hydraulic piston, rack and pinion, and the power screw. The hydraulic piston would require a complete hydraulic system and associated controls. A hydraulic piston would be advantageous for the use of a cone penetrometer since the hydraulic pressure could be measured to provide the cone loading profile. There would be no advantage in the use of this system for any other instrumentation. The cost and complexity of the associated auxiliary equipment and controls makes the hydraulic piston impractical. The horizontal dimensions of the hydraulic piston would be compatible with the design, but, in order to provide for a 5 ft stroke, the device would have to be at least 10 ft high and would extend 3 to 4 ft above the platform.

The rack and pinion has dimensional problems similar to the hydraulic piston. Due to the torque and speed requirements for the system it is impractical to have the pinion translate on a stationary rack. A translating rack would require close tolerance linear bearings in order to maintain proper mesh. Since the sea water near the ocean floor contains considerable debris (sediment, shells, micro organisms living near or on the bottom) it would be undesirable to require close tolerances. In order that the rack would not have to penetrate the sediment, the drive point would have to be located approximately 6 ft above the sediment, near the top of the main frame. This problem would not be insurmountable, but would require considerable structural framing. The device was rejected in favor of the power screw.

A threaded block on a power screw provides the required translatory motion within a minimum volume and can be used for any instrumentation (Fig. I). An American Standard I-beam, 8I6.348 is the main structural member of the device (Fig. I-5).

The beam has an 8 in. depth and a 4 in. flange width and is of 6061-T6 aluminum. An American Standard beam was chosen because there is no Aluminum Association Standard (AAS) I beam with an 8 in. depth and a 4 in. width. An AAS I beam would be desirable due to the flat ends on the flanges, but such a beam would have to be specially ordered and the amount required for this platform would not warrant the expense. Two bearing plates (Fig. I-1, I-5-1) are fixed

to the ends of the beam to support the screw. The upper plate is bolted on to allow for disassembly of the device. A square threaded screw shaft of 1020 steel (Fig. I-2) is positioned parallel to the major axis of the beam and is supported at both ends by an open race, ball, thrust bearing and a sleeve bearing of Delrin AF. Delrin was chosen as bearing material because it incorporates a bearing surface with an insulator between the steel shaft and the aluminum bearing plate (Fig. I-1-1). The block for mounting instruments (Fig. I-3) is of 6061-T6 aluminum with an insulated, threaded steel core. The block overlaps the beam to restrict its motion to translation and has a maximum travel of 4 ft. The top of the screw shaft extends above the upper bearing plate and is turned to accept a roller chain sprocket for power transmission.

Due to the fact that debris will be present in the operational environment, the minimum thread clearance is required to be 0.020 in. There are no standard dies which will cut a thread of this tolerance. The screws and instrument mount block cores will have to be turned. This is justified if the possibility of the binding of close tolerance threads is considered.

In order to design the device a determination of the suspected loading during insertion and withdrawal is required. This depends on the depth of insertion, diameter of the probe, and the cohesive properties of the sediment. The following is an empirical equation for the extraction load

for a straight cylindrical rod [3]:

$$F = 1/2\ A \cdot VS$$

where;

F = force (lb)

A = submerged area (in^2)

VS = vane shear strength of sediment (lb/in^2)

The average depth of penetration of the test instruments is 3 ft and they are not normally greater than 3 in. in diameter. The maximum vane shear strength is expected to be near 3 psi. These values yield a calculated withdrawal force of 519.5 lb. A review of 3 in. diameter cone penetrometer data supports the use of this as a design limit. With a safety factor of approximately 3, the design insertion and withdrawal load is 1,500 lb. The load will be considered as acting parallel to the axis of the screw and offset by 7 in. This allows for 2 in. between the face of the instrument mount block and the center line of the probe.

This design criteria will permit the insertion and withdrawal of a 3 in. diameter probe to a depth of 4 ft in nearly all known ocean sediments. The protection against probe damage from striking an impenetrable object, such as a rock or large clam, will be discussed in the section on motor boxes.

The drive frame is fixed to the main frame by being fabricated to fit around the pipe and clamped to the upper and lower rectangular bars (Fig. I).

Syntactic foam is molded into the voids of the drive frame I-beam forming a rectangular cross-section. The instrument drive, less motor box and chain drive, weighs 205 lb in air and 105 lb in water.

It is recommended that eight complete instrument drives be delivered, as the basic complement of the platform. An additional drive is to be provided for coring.

Provisions for insertion depths greater than 3 ft can be found in the section on alterations.

C. CORING DEVICE

The mission of the coring device is to take and store cylindrical cores of the sediments for examination and classification on board the support vessel.

There are two basic corers presently used for ocean sediments. They are the gravity corer and the piston corer. The gravity corer consists of a long hollow tube-like barrel with a weight on one end. The corer is dropped, on a cable, from a surface ship and upon impact the barrel is driven into the sediment. The corer is then lifted to the surface by the cable and the sediment sample is pushed from the barrel. The impact of the barrel with the sediment destroys the stratification of the upper portion of the sediment and the entire core is disturbed along the barrel-sediment interface.

In order to alleviate this problem, the piston corer was developed. This corer is similar to the gravity corer except that a piston is placed inside the barrel. Upon impact the piston is held at a given short distance above the sediment as the core barrel passes on into the sediment. The piston produces a low pressure area within the core tube which tends to reduce the compaction and disturbance of the upper stratification of the sediment. Piston cores have been used successfully for cores over 50 ft in length. In our design the piston-type coring concept is incorporated into each of the core barrels.

The core barrels are made from 3 x 1/4 in. steel pipe which is tapered on one end to provide a cutting edge. The pipe is split lengthwise and removable pin piano hinges are welded to the pipe where it has been split in order to provide a facility for removing the core with a minimum of disturbance (Fig. II-3-1). A piston is placed in each core barrel and is held approximately 1 ft above the sediment by a cable attached to the top of the core cylinder (Fig. III-3-5, III-6-2).

Since the core barrel can be considered as a probe of the same dimensions used to design insertion and withdrawal, an instrument drive is suitable for providing the mechanical power required to take a core. The main frame is designed to accommodate an instrument drive in position for coring. A connector plate is required to link the instrument drive and the core barrel (Fig. III-4).

In order to provide for a multiple coring capability the device must be capable of linking an empty core barrel to the instrument drive and store filled core barrels for retrieval at the surface. Samuel Colt's invention finds application here. The simplest device is a rotating cylinder carrying core barrels to and from the instrument drive link by incremental rotation (Fig. III-1, IV-2-8). The core barrels are contained in tubular guides which are slotted to allow for movement of the lifting bearing block (Fig. III-1-3, III-3-2). The lifting bearing supports the core during stowage and provides the link which transmits the force during coring. During stowage, the lifting bearing rides on the core carrier ring which is welded to the main frame (Fig. III-5-1, III-6-1). When a core barrel is in position for coring, the lifting bearing fits into a slot in the instrument drive-core connector (Fig. III-6-2). During coring the core barrel is moved downward within the core guide and is centered prior to insertion by the combined action of the instrument drive-core connector and the lower core guide (Fig. III-2-6, III-4-1).

The core cylinder is supported at the top by the main frame and at the bottom by the ballast tank and core cylinder support frame. The support frame clamps to the lower main frame, to provide for ease of assembly and transportation. The bearing surfaces are of Delrin AF.

The void areas in the core cylinder are filled with cast syntactic foam to provide additional buoyancy. The air

weight of the core cylinder, eight cores, support frame, and empty ballast tank is 1,900 lb. The weight in water with the ballast tank flooded is -230 lb.

In order to insulate the steel core barrel from the 6061-T6 aluminum core guide, a thick coat of silicon grease should be applied to the outside of the barrel and lifting bearing assembly, prior to loading into the core cylinder. The bolts are insulated from the frame clamp by Delrin washers and plastic electrical tape. The electrical tape is to be wrapped around the bolt shank to a diameter of 0.60 in. The frame clamp requires 20, 1/2-20 UNF bolts with 1 1/2 in. shank. The bolts are of K-Monel.

D. BALLAST TANK

The ballast tank is integrated into the design of the coring cylinder. The tank consists of a 6061-T6 aluminum cylinder 24 in. in diameter, 1/2 in. thick, and 82 in. long with a 1/2 in. plate closure at the top. The tank is supported by a circular plate which is welded to the cylinder, 12 in. above the bottom of the tank, and to the frame clamps (Fig. III-2).

A blow and vent connection manifold is welded into the top plate. Air will be supplied to the tank, during blowing, at 15 psi above the ambient pressure. The bottom of the tank is open to allow for free movement of water into and out of the tank.

The portion of the tank near the support plate provides an axial bearing surface for the core cylinder. The upper portion of the tank is free, allowing for expansion during blowing.

The ballast tank provides ± 415 lb buoyancy at a depth of 8,000 ft and, ± 650 lb near the surface. The variation is due to the weight of the air in the tank. It should be noted here that the maximum amount of usable air that can be carried to 8,000 ft is approximately 400 lb. This arises due to the maximum available negative buoyancy in the ballast tank. If 400 lb, by weight, of air is used at 8,000 ft the platform will become neutrally buoyant with the ballast tank flooded. The air flask volumes required, based on this, are 52.5 ft^3, 17.34 ft^3, and 11.99 ft^3 for 5,000 psia, 8,000 psia, and 10,000 psia service pressure respectively.

The 400 lb air loss requirement restricts the usable volume of air at 3,520 psig to 38,150 in^3, a little over the amount required to blow the ballast tank dry at 8,000 ft. This is restrictive within the context of the platform's desired operations, but with prudent operation it will provide adequate amounts of buoyancy variation. If the volume of air available is restricted to the values given and the pressure at launch is as required, the system has a fail safe nature in that as the flask approaches sea pressure at 8,000 ft, the platform approaches neutral buoyancy with the ballast tank flooded. The platform would be retrievable by the submersible even if the usable air in the flasks was

depleted (i.e. flask pressure equals sea pressure). As the platform is moved from the ocean floor, any air in the ballast tank, service piping, and reservoir flask will expand and add to the buoyancy of the platform. After a short distance, the platform should be able to continue on its own.

A low initial flask pressure will cause the platform to be negatively buoyant at flask depletion. This would cause serious problems and should be avoided. The following equations should be used to determine the required flask pressure from the pressure at operating depth and the reserve buoyancy:

$$P_f = \frac{P_s(B/SW)}{V} + P_s$$

where:

P_f = pressure (psia) of air flask at 0°C - must be corrected to atmospheric temperature

V = volume of air flask (in^3)

P_s = sea pressure at operating depth (psia)

B = actual, initial, negative buoyancy from compensation (lb)

SW = specific weight of air at operating depth ($lb/in.^3$) given by:
$SW = 3.1745 \times 10^{-6} (P_s/c) + 4.671 \times 10^{-5}$
from modified perfect gas law assuming isothermal expansion, "c" is compressibility factor

The volume of air expended prior to flask depletion is given by $V_d = B/SW$.

Since the resulting pressure in the flask depends on the reserve buoyancy which in turn depends on the pressure in the flask, the process must be iterative. This procedure should be performed to determine the required flask pressure prior to any operation at any depth. The iteration consists of selecting a reasonable reserve negative buoyancy and calculating a required flask pressure. This pressure, in turn, is used to determine the reserve negative buoyancy by use of the compensation program. An agreement of ± 10 lb in the reserve buoyance is sufficient.

Further discussion of the air system is provided in Section IV-H.

E. MOTOR BOXES AND DRIVE SYSTEMS

The motor boxes are oil filled, pressure compensated housings for the electric motors and gear reductions. The drives are prime movers for the instrument drives, rotation of the core cylinder, and the pumps for hold down suction. The boxes are constructed of welded 1/4 in. 6061-T6 aluminum plate with a removable cover to allow access to the internal components. The framing required to support the motor and various shaft bearings is integrated into the fabrication of the boxes by welding 1/4 in. plates of 6061-T6 aluminum to the interior. Small extensions of pipe are welded to the housing to allow for the fitting of the electrical conductor tubing and the pressure compensation diaphragm.

The large voids, primarily around the motor, are filled with molded syntactic foam blocks and the remainder of the voids are oil filled. The oil to be used is normal low additive lubrication oil of SAE 10 weight. Oils of this type have a bulk modulus of elasticity of approximately 2.7×10^5 psi. This results in a volume change of 1.3% at 8,000 ft. The volume of oil used must be minimized in order to limit the size of the compensation diaphragm.

The specified fasteners for the cover are of molded nylon. This eliminates the need for insulating steel bolts and will provide adequate strength to seal the cover. A silicone sealant should be placed on the bolt flanges to seal the box prior to filling with oil.

The motors chosen for the drives are 24 Vdc with permanent magnet fields. These newly developed motors provide extremely high torque at very low current. The weight is approximately a third of a comparable wound field dc motor, and the displaced volume is approximately one quarter. In general dc motors, submerged in oil at sea pressure, have been extensively used in deep ocean work. The only drawback is in the reduced brush life. This is not expected to be a problem due to the short operation time between service availabilities.

The pressure compensation devices are rolling diaphragm air cylinders which can be adapted by removing the internal return spring. This provides for movement of the piston with zero pressure differential across the diaphragm. The

normally pressurized side of the diaphragm is oil filled and sea water is allowed to enter the cylinder vent. The shaft, extending a few inches from one end of the cylinder, is totally unloaded and can be used to visually determine the amount of volume change in the oil during operation of the platform.

1. Instrument Drive Motor Box

This drive system is powered by a 3/4 HP, 1140 RPM, dc motor. A single stage gear reduction, internal to the box, produces an output speed of 190 RPM at full load. The screw shaft of the instrument drive is powered through a chain drive providing a further reduction in speed to 119.5 RPM at full load. It is recommended that a torque limiter replace the small sprocket. This will provide insurance against over driving an instrument during insertion. A torque limiter setting of 244 in-lb will provide an insertion-withdrawal force of 1,500 lb (Appendix A, Sect. A). The recommended setting is 220 in-lb. The torque limiter must be set while submerged in sea water (Fig. IV-1).

The recommended pressure compensation device is Bellofram Corp. #S-4-F-BP-CFM less return spring. The device can be mounted anywhere on the instrument drive and connected to the motor box by 2 in. ID extruded vinyl tubing. This device is capable of compensating for 7.2 in^3 of volume change.

2. Core Cylinder Motor Box

This drive system is powered by a 1/12 HP, 2500 RPM, dc motor. A four stage gear reduction, internal to the box, produces an output speed of 1.93 RPM. The core cylinder is rotated through a chain drive providing an additional reduction ratio of 8:1. A single rotation of the output shaft of the motor box properly aligns a new core tube with the core insertion drive. A torque limiter is provided on the bull gear of the fourth stage of the reduction. A torque limiter setting of 900 in-lb will provide protection against gear damage in the event of cylinder lock-up. This limiter must be set while submerged in oil. The limiter can be set as low as 300 in-lb without impairing the operation of the drive (Fig. IV-2, Appendix A, Sect. C).

The recommended pressure compensation device is Bellofram Corp. #S-16-F-BP-CFM, less return spring. The device can be mounted anywhere near the core drive and connected by 2 in. ID extruded vinyl tubing. The device is capable of compensating for a 67.2 in^3 volume change.

3. Motor Box for Hold Down Pump

The fabrication of this motor box has not been explicitly defined herein and is left to the imagination of the construction facility. A dc motor is to be used to drive a reversible vane pump to provide hold down suction and lift off pressure to the hold down plates. The pump should have an inlet diameter not less than 3/4 in. and be capable of sustaining a zero flow head of at least 15 psi.

This will require approximately 1/8 HP. The motor is to be enclosed in an oil filled, pressure compensated housing. A short section of 6 in. ID, 6061-T6 aluminum pipe with end plates would suffice. A lip type press fit shaft seal, similar to that in Fig. IV-1-3, should be used. The motor box and pump are to be fixed to a 6061-T6 aluminum plate with appropriate mounting devices to allow for easy placement on the main frame. Figure A shows a sketch of the pump system.

The recommended pressure compensation device is Bellofram Corp. #S-4-F-BP-CFM, less return spring.

F. BATTERY BOX

The battery box is an oil filled, pressure compensated container for the platform's batteries. The basic box is constructed of 1/4 in. 6061-T6 aluminum plate closely conforming to the dimensions of the battery pack in order to reduce the volume of oil required. The 16 gauge, 6061-T6 aluminum cover is domed to allow for the collection of gases during charging and discharge. The attachment of the cover is similar to that of the motor boxes. A pressure relief valve is located at the top of the domed cover to allow for release of gases while submerged and on board the support ship. The recommended relief valve is Circle Seal Products no. D532A-1M-4 set at a cracking pressure of 2.5 psig. The static pressure head in air at the level of the relief valve is approximately 1.2 psig. It is recommended that the pressure compensation device be attached to the end of the box

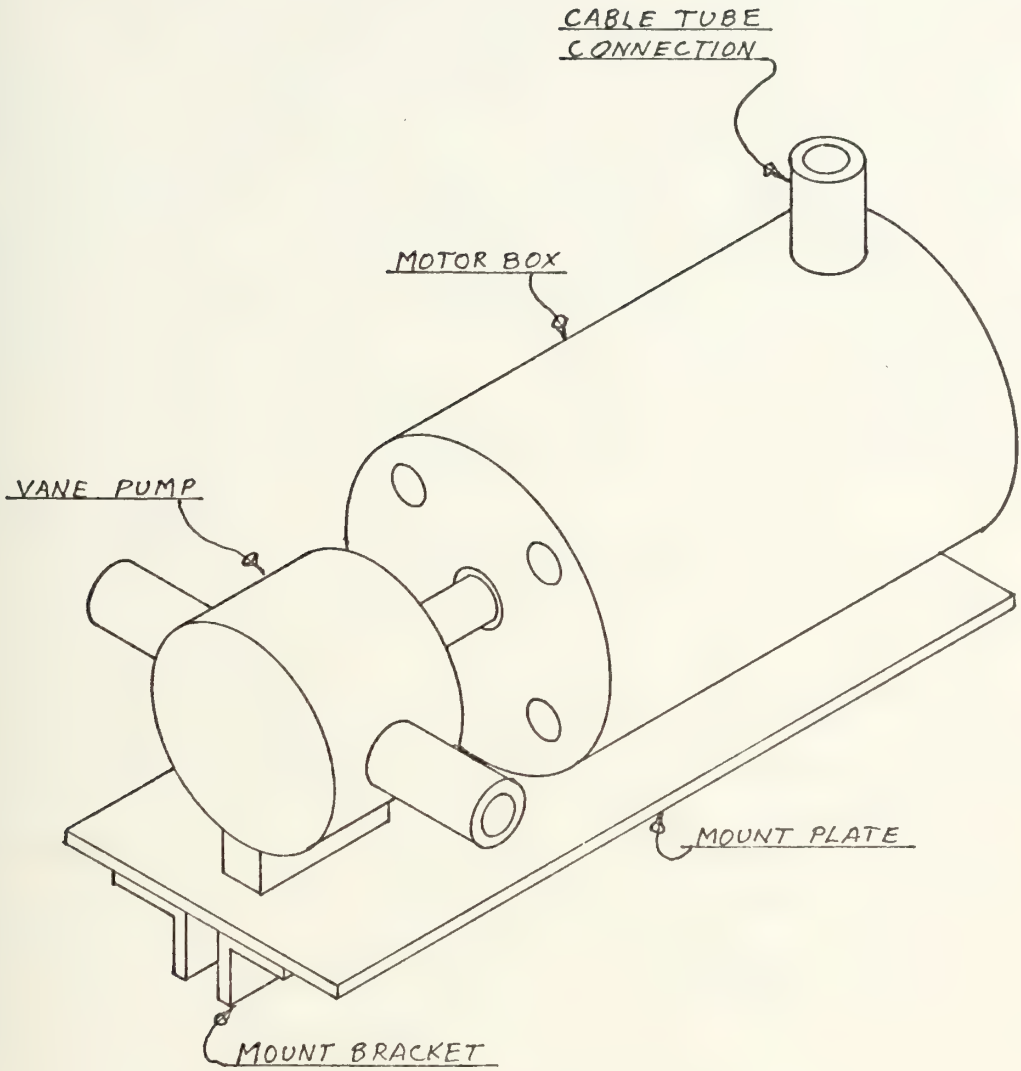


FIG. A. HOLD DOWN PUMP

such that its highest oil filled point is above the top of the cover. The recommended pressure compensation device is Bellofram Corp. #S-36-F-BP-CFM, less return spring. This device will compensate for 216 in^3 of volume change.

A connection in the end opposite the pressure compensation device is provided for the power cable tubing. The charging location is on the side of the box. The recommended connector to be mounted in the box is Electro Oceanics, Inc. no. 53-H-2-F-1. The mating connector for the charging cable is no. 53-H-2-M-1.

Figure B is a sketch of the battery box.

1. Batteries

The recommended battery is the Sears Roebuck and Co. No. 27E, sold under the brand name of "Diehard". The battery provides 12 Vdc and is rated at 4250 amp-min for a 25 amp discharge rate. This rating is used in determining the platform requirements since the expected average current requirement will be approximately 25 amp.

Eight batteries are connected in a series-parallel arrangement as shown in Fig. C. This provides a 24 Vdc service rated at 17,000 amp-min. The platform, less instrumentation, is expected to require a total of 7,792 amp-min to complete eight operations on the ocean floor. This leaves 9,208 amp-min to provide instrumentation power.

An operation is defined as the sum of the insertion and withdrawal process, performed at full design load, of each of nine instrument drives, the positioning of a new

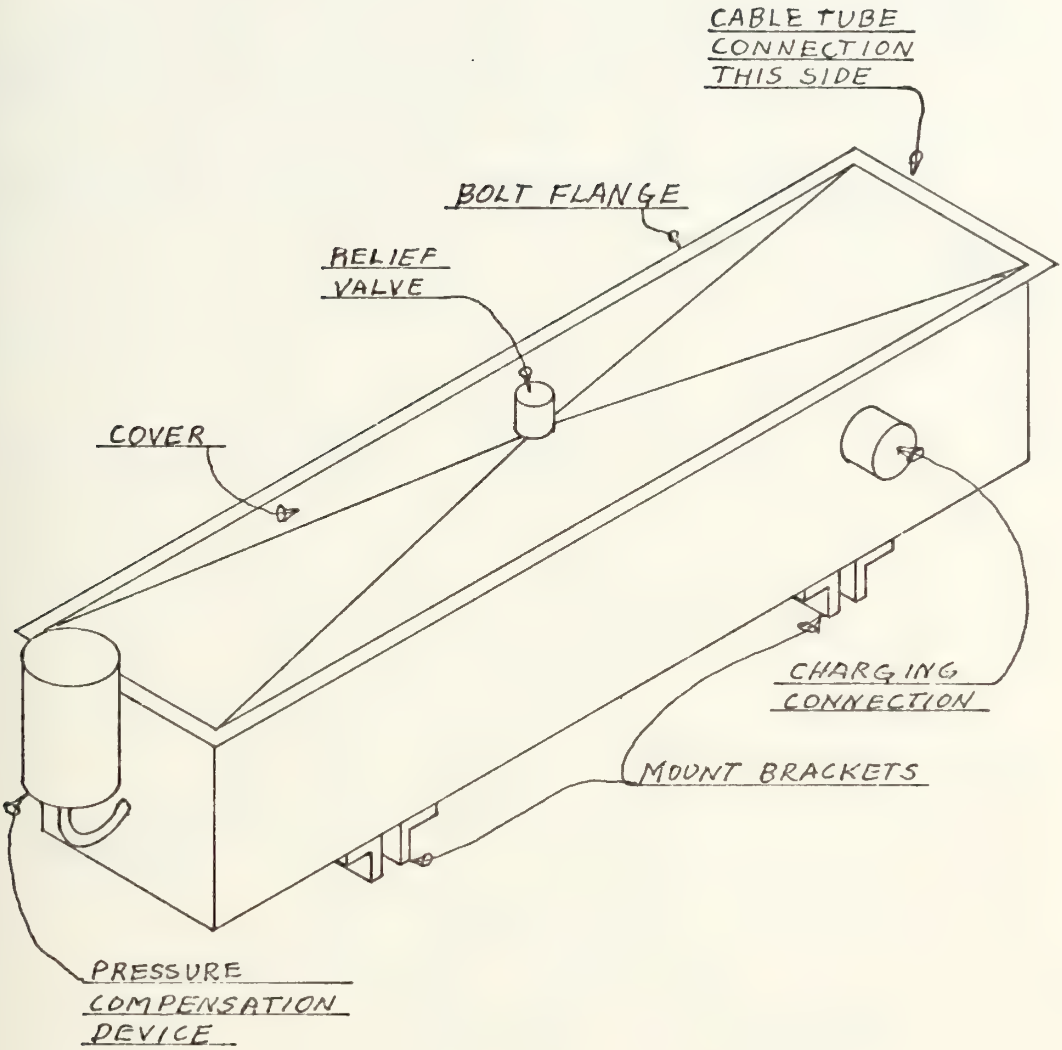


FIG. B. BATTERY BOY

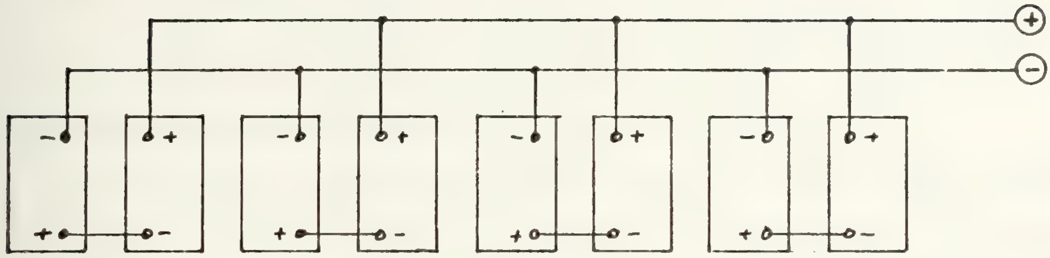


FIG. C. ARRANGEMENT OF BATTERIES IN BOX

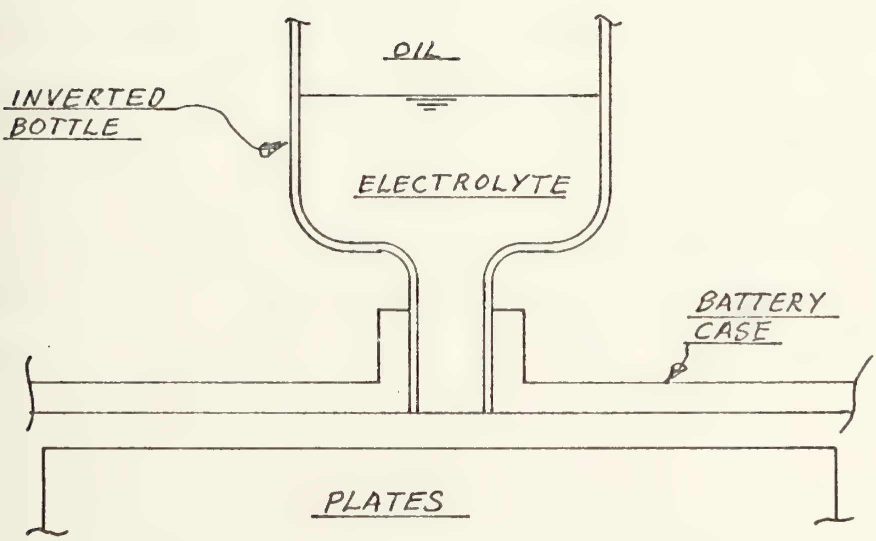


FIG. D. CROSS SECTION OF COMPENSATED BATTERY CELL

core tube by the core cylinder drive, and the operation of the four hold down pumps for a total of 18 min. Each instrument drive requires 2 min for the insertion and withdrawal process and the core cylinder drive requires 1/2 min to position a new core tube.

Pressure compensation of each cell is accomplished by fitting each cell opening with an inverted plastic bottle (approx. 50 ml size). A large hole is cut in the bottom of the bottle to allow the free communication of oil and cell gases. The cell is filled with electrolyte such that the electrolyte-oil interface is approximately at the half way mark of the bottle. Figure D is a cross section of the compensated cell.

G. CONTROL BOX AND PLATFORM CIRCUITS

The control box is an oil filled, pressure compensated container for the electrical controls of the platform and instrumentation. The design of the box is similar to the battery box, except that the cover is of flat 1/4 in. 6061-T6 aluminum plate. The recommended internal dimensions are 12 in. wide, 8 in. deep, and 36 in. long. The minimum cable tube connections required are 1 for power, 11 for motors, 5 for umbilical cables, and 4 for instrumentation. These can be conveniently located anywhere on the sides and ends of the box. During fabrication it would be prudent to provide more than this minimum to facilitate different connection configurations without alteration to the box. Any unused connectors can be easily capped.

No flotation should be used in the control box since its presence may interfere with the operation of internal components such as relays.

The cover is to be fastened to the box by molded nylon bolts. Five pressure compensation devices, Bellofram Corp. #S-36-F-BP-CFM, are mounted on the cover, providing compensation for 1,080 in^3. These devices plus the devices on the motor boxes and battery box provide for approximately four times the expected volume change. This will allow for air bubbles in the system and the addition of instrumentation without the addition of pressure compensation devices. Since tubing is used to connect all electrical devices with the control box, volume changes in other components can be compensated for at the control box.

1. Platform Circuits

The control circuits of all electrical devices on the platform consist, in general, of power relays opened or closed by pulses from the operation console in the submersible. The following is a description of the various relay circuits required and their operation.

The circuits are divided into two basic, electrically separate areas. The control circuits consist of linking the operation console in the submersible to the power relay through the umbilical cable, and general control of the power relay by limit switches. The circuits are rated at 2 amp. All limit switches open when the device is at its limit.

The second area is the power circuits. These are the simple switching of power to the motors, through relays. These circuits are rated at 15, 25, and 50 amp, depending upon the motor.

The platform requires no power unless an operation is being performed. All control and power circuits are normally open.

a. Stop Control

This circuit, Fig. E, is used to stop any and all operations. A normally closed relay is momentarily opened by a pulse from the operation console. The control circuit of all power relays is opened allowing any relay under control power to open.

b. Core Drive Control

This circuit, Fig. F, consists of a three contact, normally open, power relay controlled by a pulse from the operation console. The holding circuit consists of a limit switch in series with one of the contacts. The pulse in this case must be long enough, approximately 10 sec, to allow the cam to rotate to a position such that the limit switch is closed. A running indication light is provided on the operation console.

c. Pump Control

This circuit, Fig. G, consists of two power relays required to reverse the motor, with four contacts each. On each relay the contact sets are triple pole, single throw, normally open, and single pole, single throw, normally

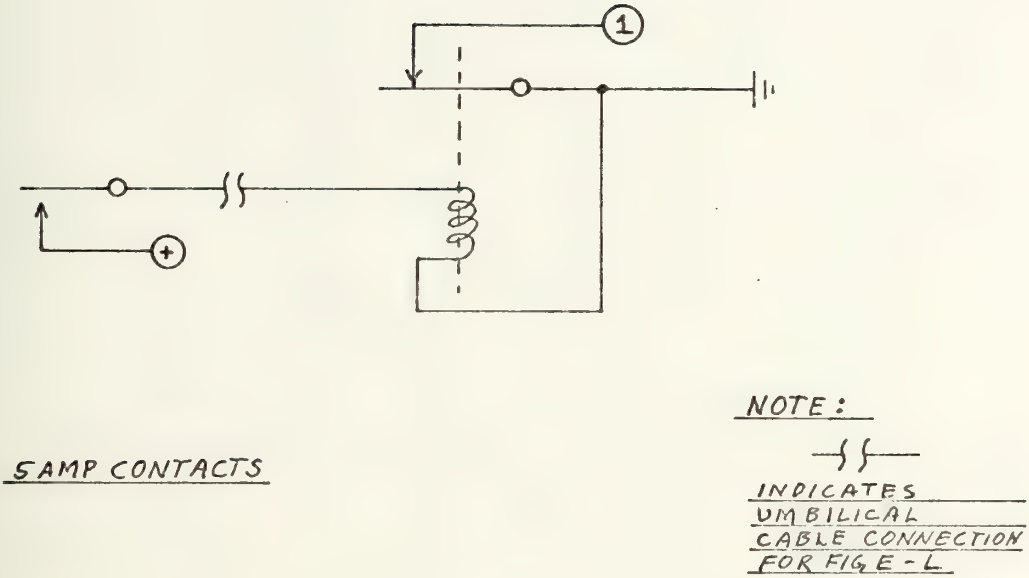


FIG. E. STOP CONTROL

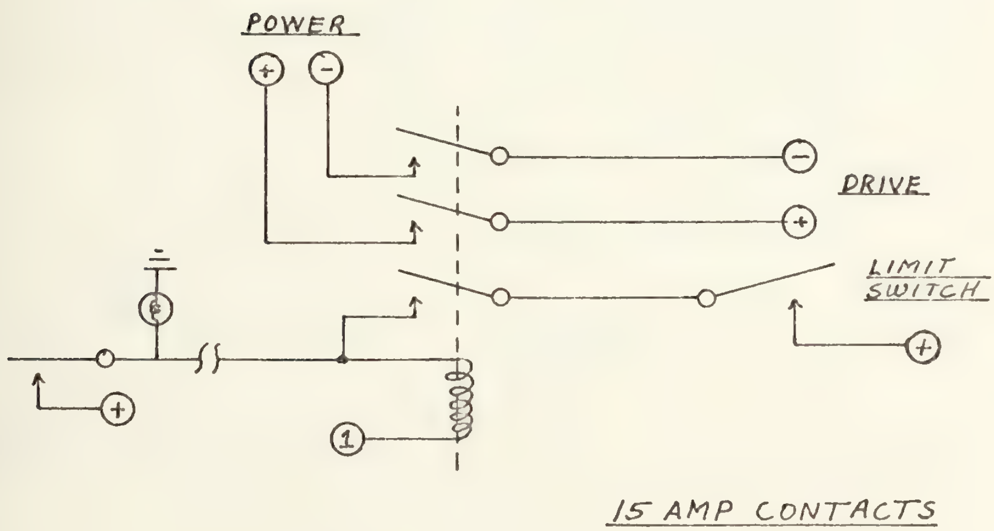


FIG. F. CORE DRIVE CONTROL

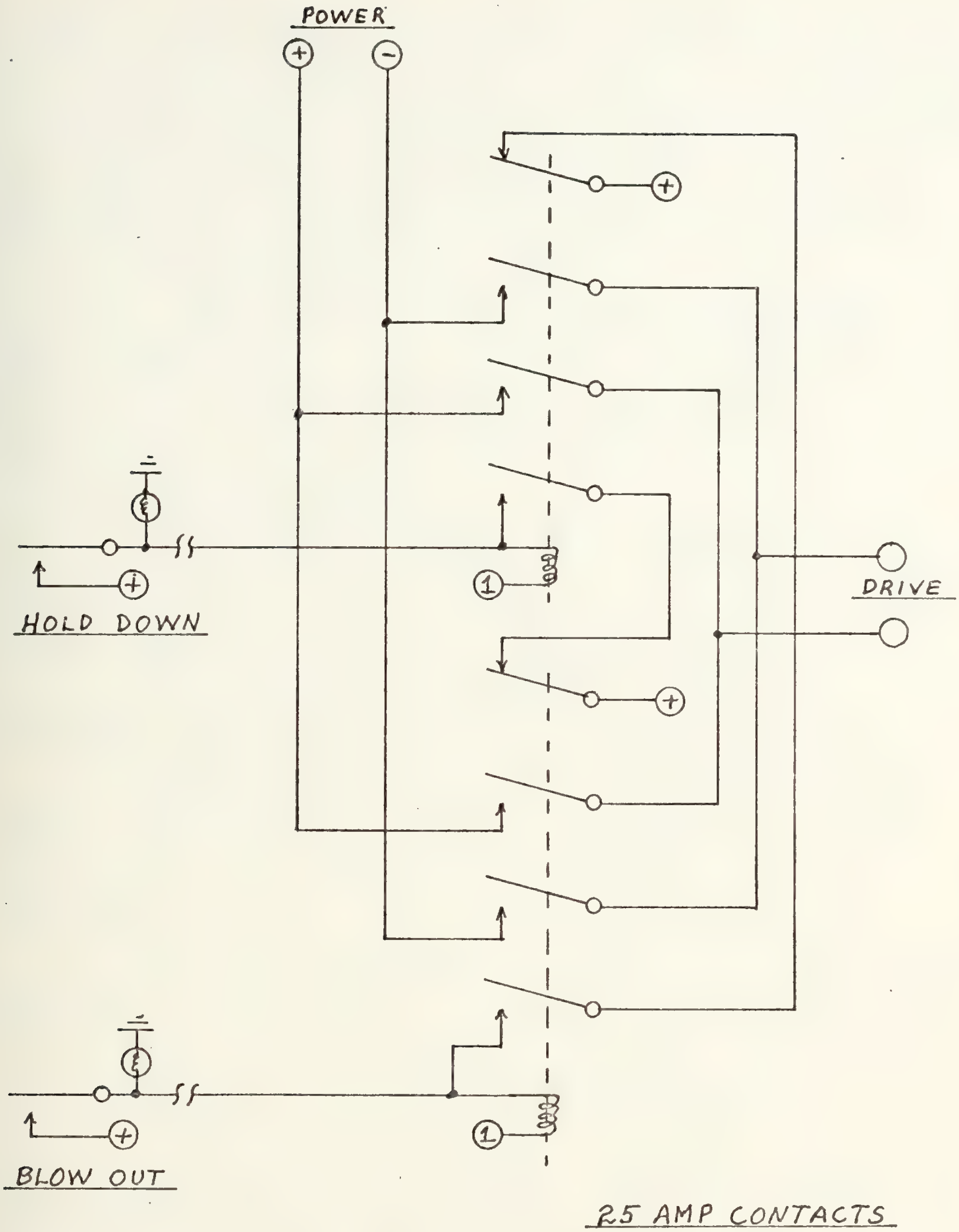


FIG. G. PUMP CONTROL

closed. Pulses from the operation console will close the desired relay to provide power to the pump. A safety circuit is provided to protect against the closing of both relays at once. The stop control must be used to stop the pumps.

d. Instrument Drive Control

This circuit, Fig. H, is exactly the same as the pump control circuit except that limit switches have been added to stop the drive at end of travel.

2. Electrical Cables

All electrical cables on the platform are contained in oil filled extruded vinyl tubing. Since this can be considered as conduit, tees and junction boxes can be used as long as the integrity of the system is maintained.

The recommended wire is vinyl covered stranded copper. Vinyl has an excellent resistance to the oil environment.

The umbilical cable is made up of five, 30 ft lengths of Electro Oceanics, Inc. no. 51-F-8-F cable. The plugs are sealed and isolated from the sea. The submersible cable is no. 51-F-8-M. Each cable contains eight conductors and is capable of withstanding 20,000 psi without breakdown. The cables are to be sealed to the control box by clamping a short section of vinyl tubing to the cable and to the box fitting.

The platform requires 27 leads to the submersible leaving 13 leads for instrumentation.

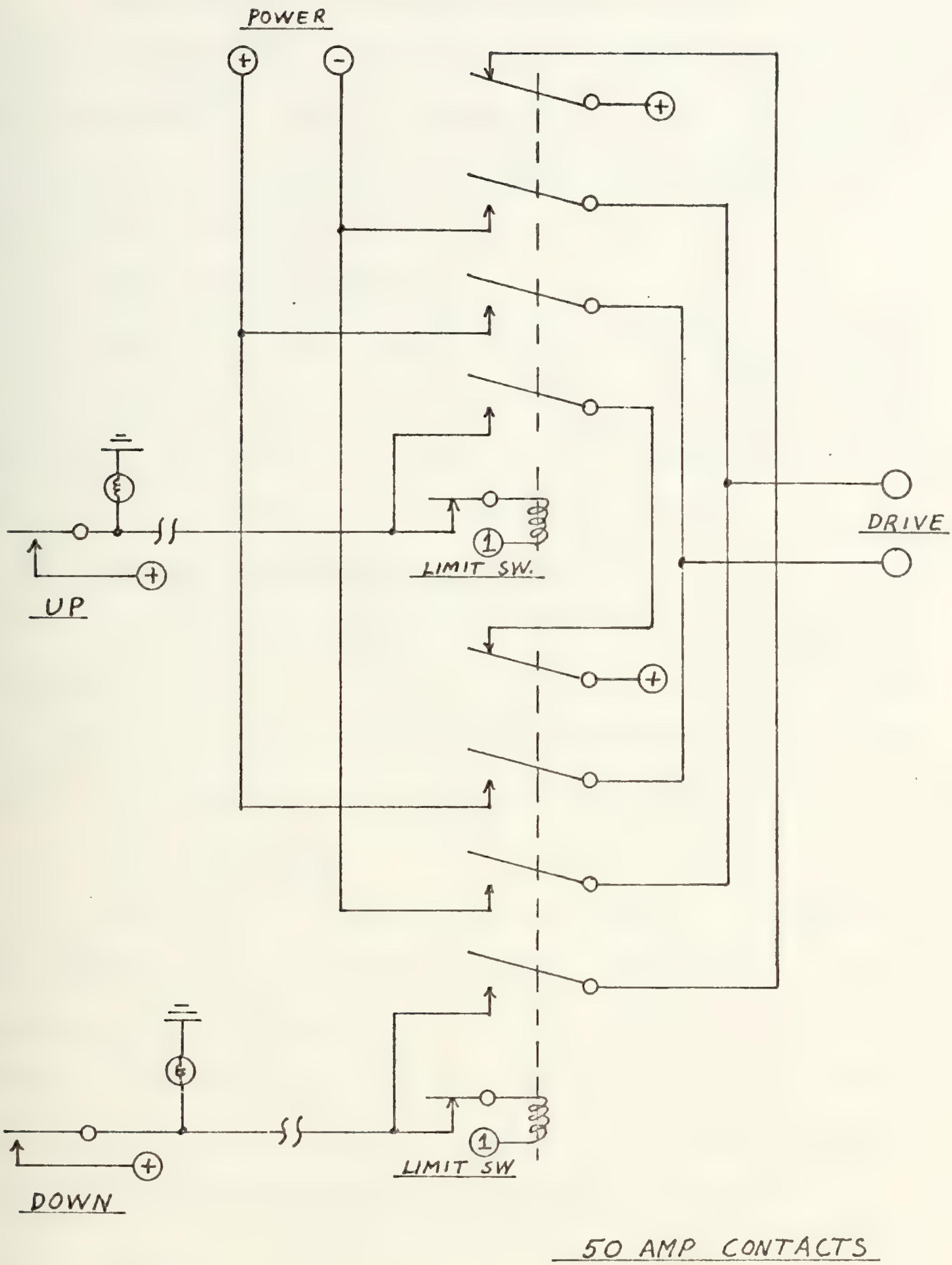


FIG. H. INSTRUMENT DRIVE CONTROL

The design of the actual mating device has not been completed since each submersible has individual characteristics and requirements. Once a submersible is selected, close coordination between the operators and the fabrication facility is required.

3. Power Circuit Breaker

Fig. J is the circuit breaker for the power circuit. The breaker, Cutler-Hammer no. SM 600 BA 100 N1, is remote controlled and rated at 100 amp. The contacts can be opened or closed by pulses from the operation console.

4. Relay Recommendations

Due to the excessive costs of special relays it is recommended that the Cutler-Hammer lightweight power relay no. SM15AXD1 be used exclusively. The power contacts are rated at 60 amp with 5 amp auxiliary contacts.

5. Limit Switches

The limit switches, Cutler Hammer no. SS12ET30-102L4, are housed in an oil filled, pressure compensated 6061-T6 aluminum case. Figure K is a sketch of a typical limit switch. The roller actuated switch is operated by movement of the spring loaded shaft. A tappet on the end of the shaft makes contact with the device to be limited.

H. AIR SYSTEM

Figure L is a diagram of the platform air system. Eight high pressure air flasks provide ballast blowing air at 15 psig, relative to sea pressure, through a bank of five staged

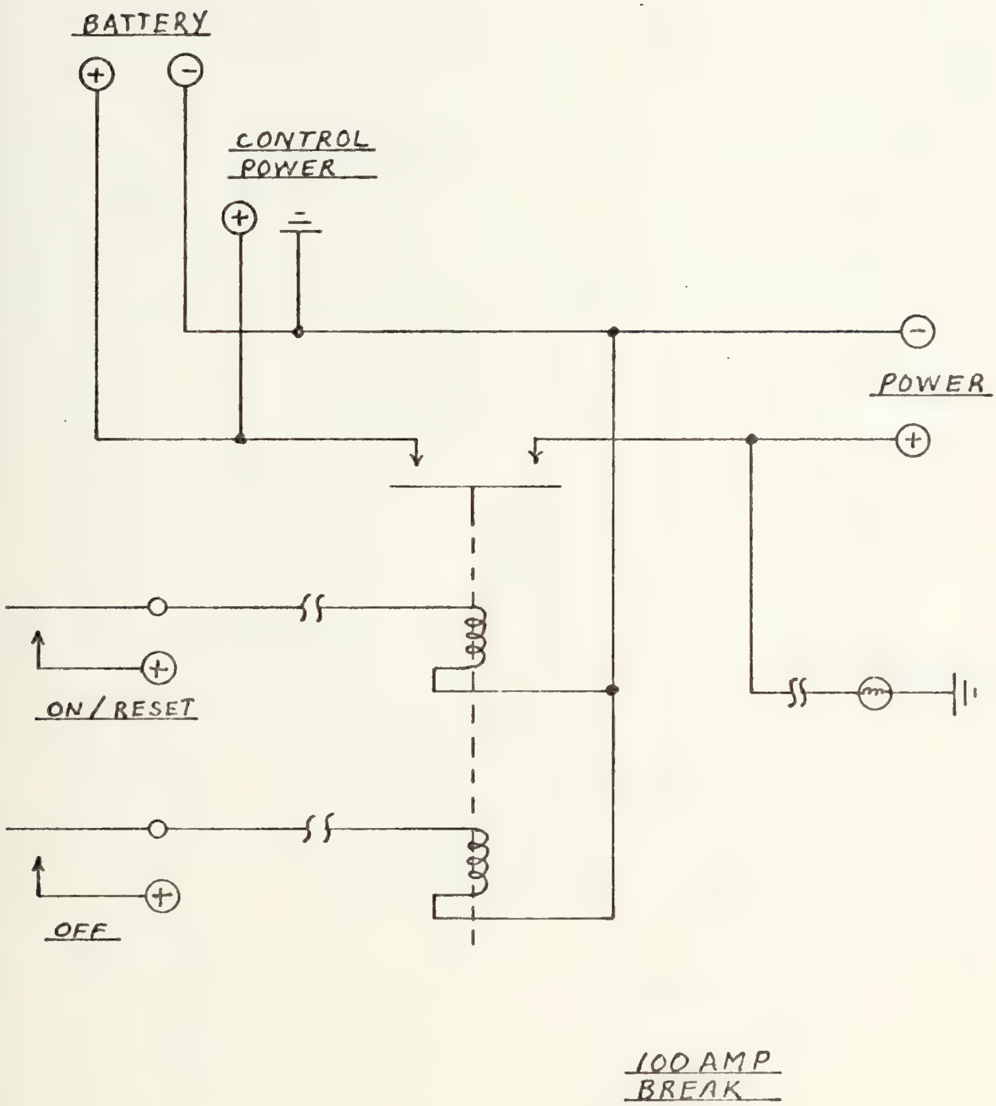


FIG. J. CIRCUIT BREAKER

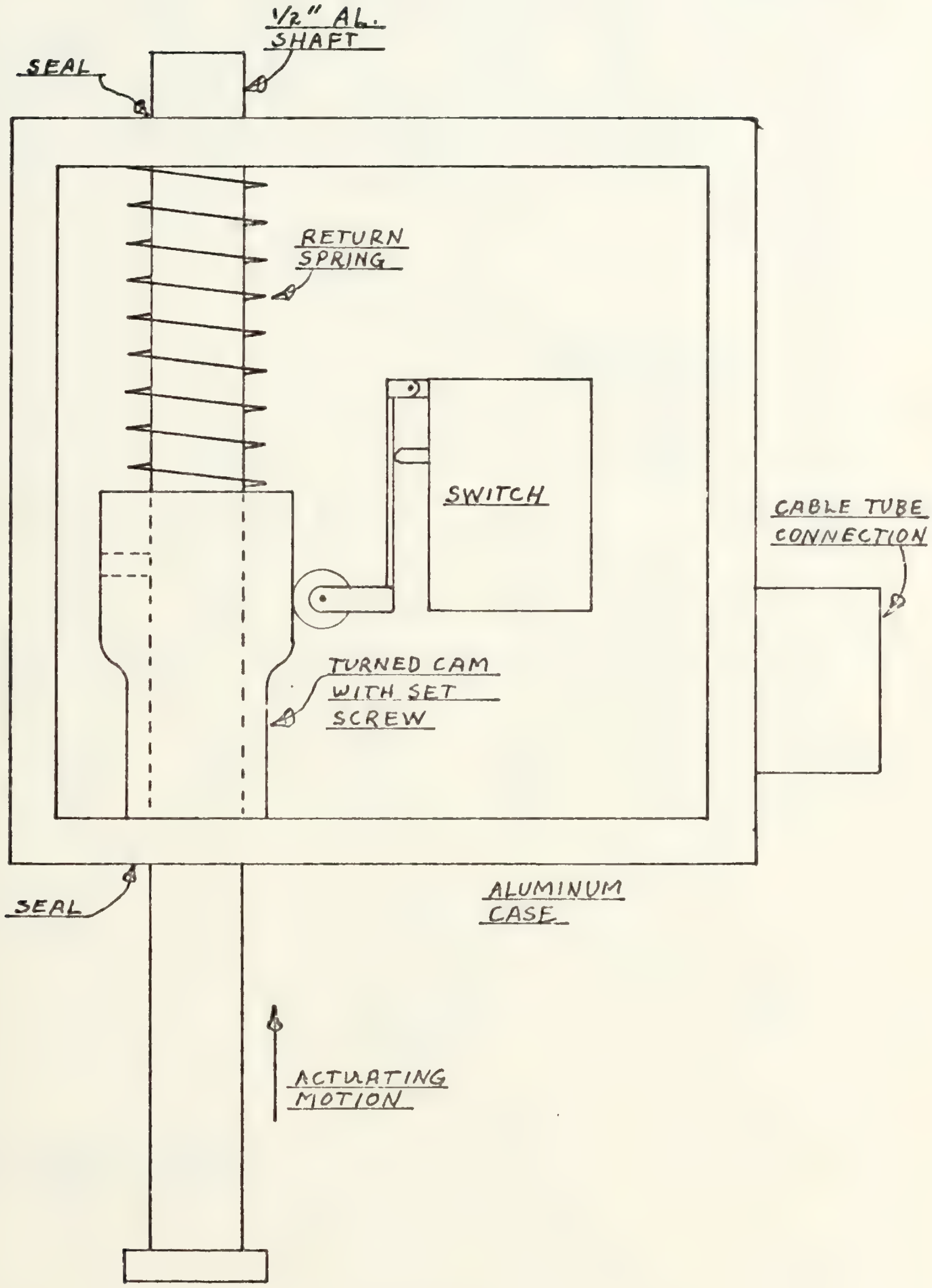


FIG. K. SKETCH OF LIMIT SWITCH

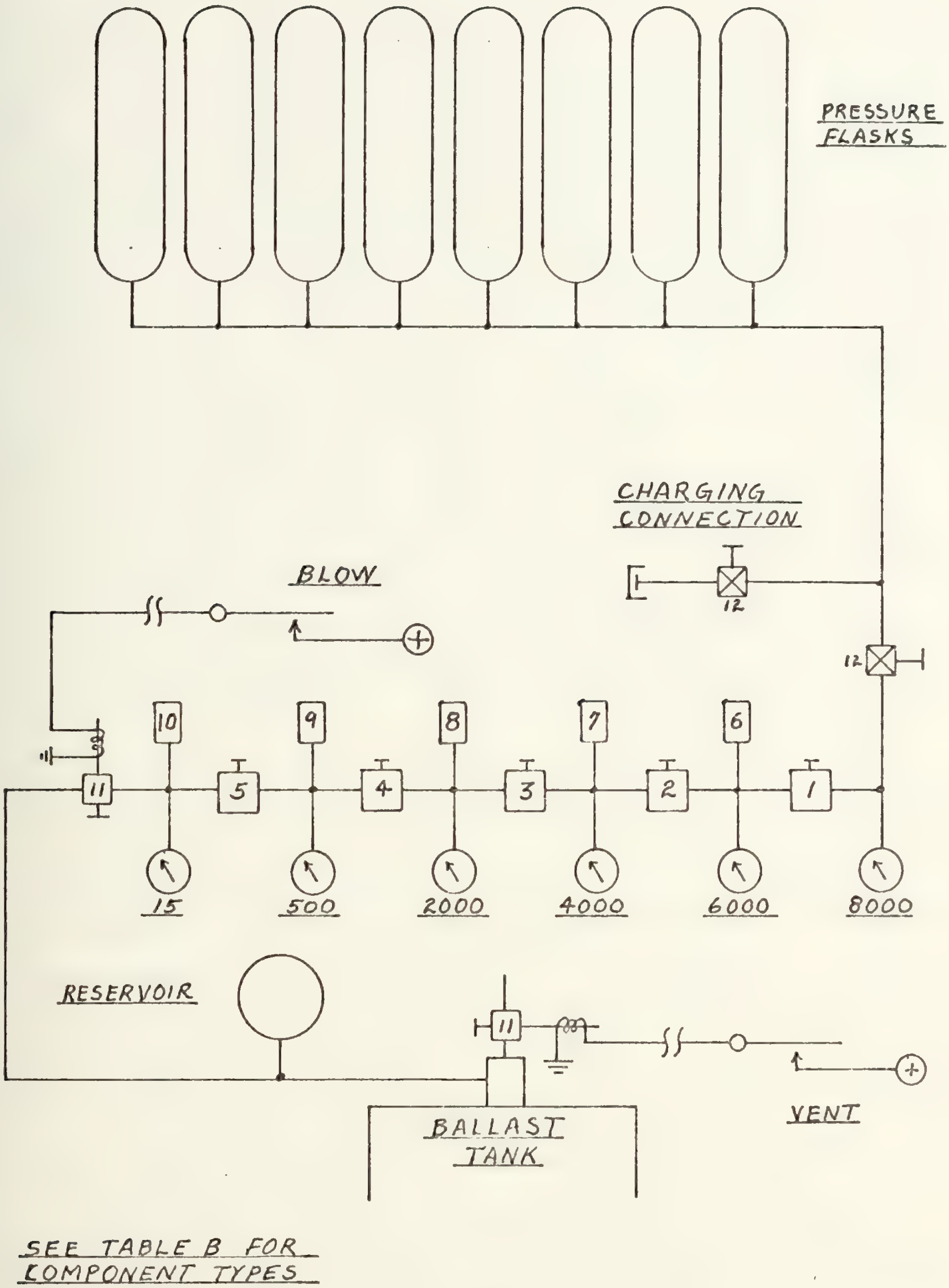


FIG. L. AIR SYSTEM

pressure regulators. Table B is a listing of component part numbers.

The high pressure air flasks are produced by National Tube Div., U.S. Steel Corp. Each has an operating pressure of 10,000 psi and a capacity of 2.50 ft^3, water volume, providing a total volume of 20.0 ft^3. The flasks are attached to the uprights of the main frame (Fig. II-2). Four pneumatic bottle clamps, Aeroquip no. MB9914-1200-S-V, are used to mount each flask to the upright. The mount plate of each clamp can be bolted to the upright. Adequate insulation between the aluminum pipe and stainless steel pipe must be used to prevent corrosion. It is recommended that the pipe be drilled and tapped and that 6061-T6 aluminum bolts be used.

The air flasks are connected with 303 stainless steel pipe with an inside diameter of 0.25 in. This pipe is also used throughout the five stage regulation bank. Aeroquip hose no. 1501-10 is used for all 15 psi service. Stainless steel reusable fittings are recommended.

The reducer bank and pressure relief valves are housed in an oil filled, pressure compensated case. Since this case is not connected to the control box it requires a pressure compensation device. Bellofram Corp. no. S-9-f-BP-CFM will provide for 27 in^3 of volume change.

Each reducer has ports for inlet and outlet pressure gauges. Regulator 1 will require both gauges. Regulators 2 through 5 require gauges only on their outlet ports. The

TABLE B

AIR SYSTEM COMPONENTS

	TYPE	MFG	NO.	NOTES
1.	Regulator	Circle Seal Corp.	GD710-S-3-3-2-G	8000-6000 psi
2.	"	"	GD710-S-3-3-2-G	6000-4000 psi
3.	"	"	IR12-250-G	4000-2000 psi
4.	"	"	IR11-250-G	2000-500 psi
5.	"	"	SR151B-S-2-2-2-G	500-15 psi
6.	Relief valve	"	5349T-2PP-6500	6500 relief
7.	"	"	5349T-2PP-4250	4250 relief
8.	"	"	5349T-2PP-2250	2250 relief
9.	"	"	5349T-2PP-550	550 relief
10.	"	"	5559A-2PP-20	20 relief
11.	Solenoid valve	Magnatrol Valve Corp.	LV-18N40	Lever override Normally closed
12.	Shutoff valve	various	-	Ball Valve, 1 in.

inlet ports of these regulators can be used for the pressure relief valves. In this manner, there are no breaks in the high pressure piping between the regulators. A 1/4 in. hole drilled into the back of the gauge cases will provide pressure compensation within the gauge. With case cover off, the gauges can be read and the regulators and relief valves can be adjusted. The relief valves are vented outside the case. The pipe penetrations in the case can be sealed by clamping a short section of vinyl tubing to the pipe and case connection.

The solenoid valves are provided with a housing which can be oil filled and a fitting to connect the cable tubing. The valves are both equipped with lever type manual overrides. By means of a lanyard attached to the lever of the vent valve, a diver can vent the ballast tank in preparation for the platform's descent to the ocean floor. Similarly, the submersible can blow the ballast tank after bottom operations are completed.

The blow valve should be mounted conveniently on the main frame and the vent valve is mounted directly to the blow and vent block (Fig. III-2-2).

The reservoir is a standard ICC pressure bottle rated at 100 psi, with a capacity of 1 ft^3. It is connected to the blow line and can be conveniently located anywhere on the main frame. The purpose of the reservoir is to damp out the shock waves caused by the sudden opening and closing of the blow valve. The reservoir also provides a volume of air at

sea pressure which can expand into the ballast tank as the platform rises from the ocean floor.

I. HOLD DOWN SYSTEM

Since the ballasting system is not capable of providing the negative buoyancy necessary to resist the force of inserting instruments, a positive hold down system is required. Numerous devices have been used to hold objects to the ocean floor. The screw and anchor spade were considered, but were found to be undesirable. The screw device consists of four large augers which would be attached to the corners of the platform and rotated into the sediment by motor drives. This would require expensive and complicated drive systems. The anchor spade consists of a shovel-like device at the end of a long pivoted arm. The device operates much the same as a fluked anchor and would be more expensive than the screw device. Both devices would be cumbersome and dangerous during shipboard operations. Both devices require separate pedestals for platform support. A differential pressure plate is used because of its low cost and simplicity in operation.

The differential pressure plate, as used on DOSP [1], consists of a flat plate which rests on the sediment and supports the platform (Fig. V-1). During hold down a vane pump is used to provide a low pressure area beneath the plate. The differential pressure across the plate provides a resisting force. The pump can be reversed to provide a blow out force, in the event that the platform becomes stuck to the sediment.

A skirt around the circular plate is provided to allow for a less than zero gauge pressure at the edge of the plate. Three or four such plates can be used to provide hold down for the platform.

The pressure distribution beneath the plate hold down can be approximated by hydrodynamics. The flow is assumed to be that of a two dimensional sink. The pressure distribution around the sink is integrated over the area of the plate resulting in the following equation for the hold down force per plate.

$$F = 2\pi[\frac{A}{2}(r_1^2 - r_0^2) + B \ln \frac{r_0}{r_1}]$$

where:

$$A = \frac{B}{r_1^2} - P_1$$

$$B = \frac{(P_0 - P_1)r_1^2 r_0^2}{r_1^2 - r_0^2}$$

and:

F = force on plate; negative value indicates a hold down force. (lb)

P_0 = pressure at suction port of the plate. P_0 is positive for negative gauge pressures. (psig)

P_1 = pressure at edge of plate. P_1 is positive for negative gauge pressures. (psig)

r_0 = minimum radius at which P_0 exists. (in.)

r_1 = radius of plate at skirt; point at which P_1 exists. (in.)

The equation can be simplified by assuming that P_0 exists only at a point, $r_0 = 0$. The equation then reduces to that of a plate with a uniform pressure distribution of P_1. With this simplification the designed plate, Fig. V-3, with $P_1 = 3$ psi provides a hold down force of approximately 1,400 lb.

A high value of P_1 indicates a high vane shear strength of the sediment and corresponds to high insertion loads (Sect. IV-B). The exact pressure distribution beneath the plate is unknown and will have to be determined experimentally. Figure V-4 and V-4-1 exhibit the test plate constructed by the shop facilities of the Naval Postgraduate School (NPS). An initial test was performed in the sediment tank at NPS. The sediment was pure Kaolinite which has a vane shear strength of nearly zero. The pressure profile exhibited was, as expected, nearly zero. Time restrictions did not allow for further testing in ocean sediments of various shear strengths. This should be done in order to more accurately determine the hold down force of the plates. The feasibility has been proven by its use in DOSP [1].

The hold down plates can be clamped to any member of the lower main frame, providing for flexibility in use.

J. FLOTATION AND MAIN FLOAT

Syntactic foam has been designated throughout the design as the flotation material. This foam is composed of small (150 micron diameter) hollow glass spheres uniformly distributed in a curable polyester matrix. The recommended product is Eccofloat no. PP32, produced by Emerson & Cuming, Inc. In the uncured state, the syntactic foam can be packed into molds and voids like damp sand. After air curing at room temperatures, the polyester matrix becomes rigid and the foam has a specific weight of 32 lb/ft^3.

This weight is half the specific weight of sea water which provides that a block of this syntactic foam weighing 1 lb in air will displace 2 lb of sea water, yielding a net positive buoyancy of 1 lb.

1. Main Float

The main float, shown in Fig. M, consists of discs of syntactic foam clamped between two 3 ft diameter 6061-T6 aluminum discs on a 4 in. diameter solid aluminum shaft. The shaft is 10 ft long with cable attachments on both ends. It is recommended that the syntactic foam be molded into discs 3 or 4 ft in diameter and 5 in. thick with a 4 in. diameter hole in the center. A number of these discs can be used to make up various buoyancy requirements. Table C can be used to determine the number of each disc to be used. The float has a capacity of 3,200 lb of positive buoyancy with twenty 4 ft diameter discs. Each 4 ft diameter disc provides 166.4

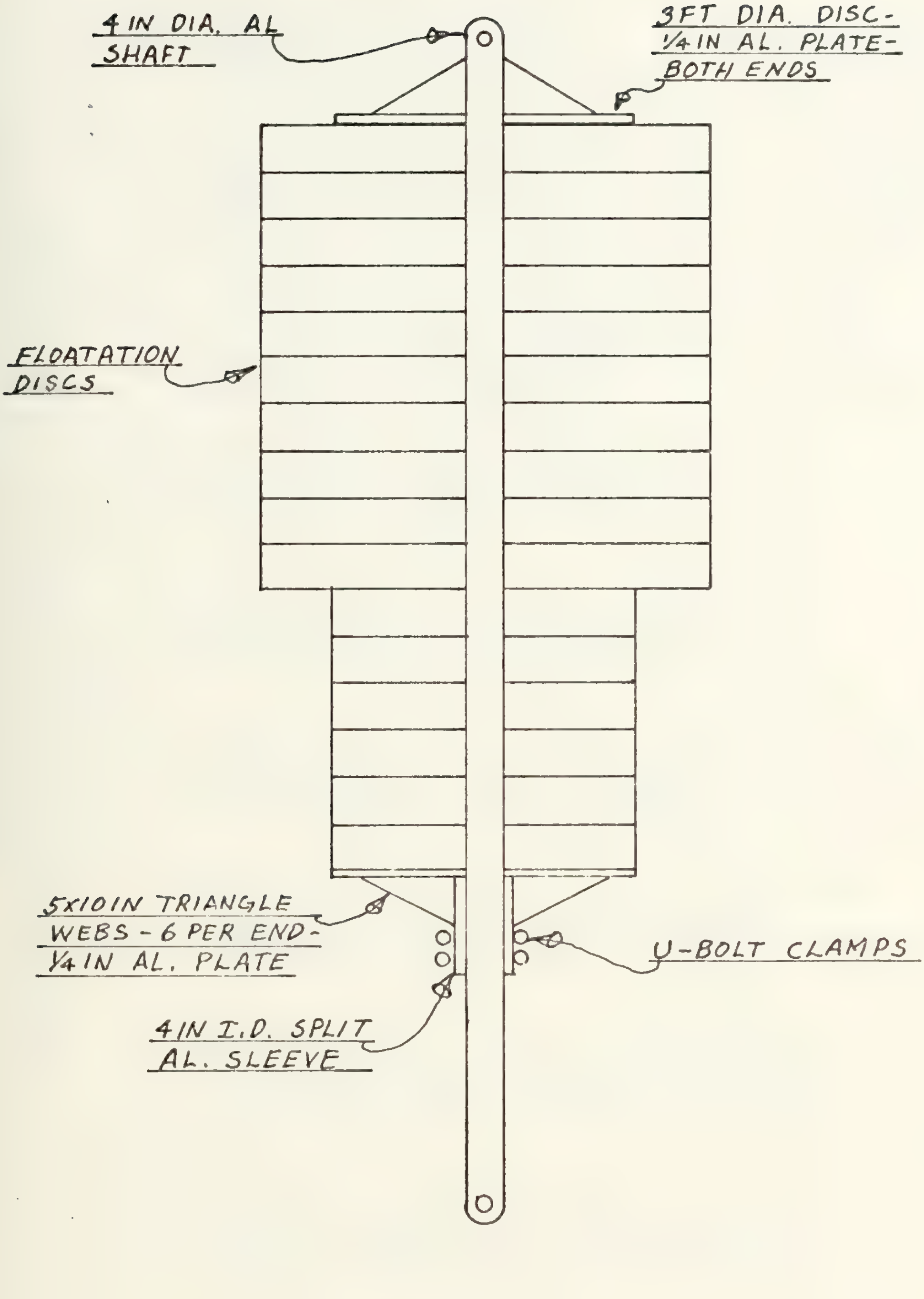


FIG. M. CENTERLINE CROSS SECTION OF MAIN FLOAT

TABLE C

FLOTATION WEIGHT, IN POUNDS

NO. OF 4 ft DISCS

NO. OF 3 ft DISCS	0	1	2	3	4	5	6	7	8	9
0	0.0	166.4	332.8	499.2	665.6	832.0	998.4	1164.8	1331.2	1497.6
1	93.1	259.5	425.9	592.3	758.7	925.1	1091.5	1257.9	1424.3	1590.7
2	186.2	352.6	519.0	685.4	851.8	1018.2	1184.6	1351.0	1517.4	1683.8
3	279.3	445.7	612.1	778.5	944.9	1111.3	1277.7	1444.1	1610.5	1776.9
4	372.4	538.8	705.2	871.6	1038.0	1204.4	1370.8	1537.2	1703.6	1870.0
5	465.5	631.9	798.3	964.7	1131.1	1297.5	1463.9	1630.3	1796.7	1963.1
6	558.6	725.0	891.4	1057.8	1224.2	1390.6	1557.0	1723.4	1889.8	2056.2
7	651.7	818.1	984.5	1150.9	1317.3	1483.7	1650.1	1816.5	1982.9	2149.3
8	744.8	911.2	1077.6	1244.0	1410.4	1576.8	1743.2	1909.6	2076.0	2242.4
9	837.9	1004.3	1170.7	1337.1	1503.5	1669.9	1836.3	2002.7	2169.1	2335.5
10	931.0	1097.4	1263.8	1430.2	1596.6	1763.0	1929.4	2095.8	2262.2	2428.6
11	1024.1	1190.5	1356.9	1523.3	1689.7	1856.1	2022.5	2188.9	2355.3	2521.7
12	1117.2	1283.6	1450.0	1616.4	1782.8	1949.2	2115.6	2282.0	2448.4	
13	1210.3	1376.7	1543.1	1709.5	1875.9	2042.3	2208.7	2375.1		
14	1303.4	1469.8	1636.2	1802.6	1969.0	2135.4	2301.8			
15	1396.5	1562.9	1729.3	1895.7	2062.1	2228.5				
16	1489.6	1656.0	1822.4	1988.8	2155.2					
17	1582.7	1749.1	1915.5	2081.9						
18	1675.8	1842.2	2008.6							
19	1768.9	1935.3								
20	1862.0									

TABLE C (CONT.)

NO. OF 4 ft DISCS

NO. OF 3 ft DISCS	10	11	12	13	14	15	16	17	18
0	1664.0	1830.4	1996.8	2163.2	2329.6	2496.0	2662.4	2828.8	2895.2
1	1757.1	1923.5	2089.9	2256.3	2422.7	2589.1	2755.5	2921.9	3088.3
2	1850.2	2016.6	2183.0	2349.4	2515.8	2682.2	2848.6	3015.0	3181.4
3	1943.3	2109.7	2276.1	2442.5	2608.9	2775.3	2941.7	3108.1	
4	2036.4	2202.8	2369.2	2535.6	2702.0	2868.4	3034.8		
5	2129.5	2295.9	2462.3	2628.7	2795.1	2961.5			
6	2222.6	2389.0	2555.4	2721.8	2888.2				
7	2315.7	2482.1	2648.5	2814.9					
8	2408.8	2575.2	2741.6						
9	2501.9	2668.3							
10	2595.0								

NO. OF 3 ft DISCS	19	20
0	3161.6	3328.0
1	3254.7	

lb of positive buoyancy and each 3 ft diameter disc provides 93.1 lb.

The purpose of the main float is to make up buoyancy of the platform and provide surface stability. The positioning of the float well above the platform reduces operational interference on the platform and provides for a high center of buoyancy relative to the center of gravity. The metacentric height of the platform and main float is approximately 20 in. Upon surfacing, after submerged operations, the platform is maintained well below the zone of wave effects for moderate seas. The platform is very stable and its pickup by a support ship would not be hampered by radical movement of the platform.

V. FABRICATION AND ASSEMBLY OF FINAL DESIGN

A. FABRICATION

The platform is fabricated primarily from aluminum - approximately 85% of all metals used. The primary fabrication technique is welding. The aluminum alloy 6061-T6 was chosen for its excellent weldability in addition to its high strength and corrosion resistance. The alloy requires no heat treatment after welding. The recommended welding method is by inert gas using a filler rod of 4043 alloy aluminum. The Aluminum Company of America produces an excellent handbook on welding aluminum and it is highly recommended as a source of technical information on the subject. The only welded steel joint is on the core barrel-lifting bearing block and the use of an iron powder rod will suffice. The preparation of surfaces for welding has not been specifically treated here since the requirements have been standardized. The handbook previously mentioned is a good source for these standards.

Bolted joints are the only other method of fastening used on the platform. These joints can be easily subdivided into two areas for bolt classification. All covers for boxes, such as motor and battery boxes, are joined by molded nylon bolts. These bolts are inexpensive and, since they are non-conductors, require no electrical isolation. All other bolted joints are made up of K-Monel bolts and

insulation. K-Monel alone has good corrosion resistance, but it must be insulated when used to join aluminum. In joints where the loading is critical, the insulators are fabricated from Delrin. All other joints can be insulated by the use of Delrin washers and plastic electrical tape wrapped around the bolt shank.

The following discussions of component fabrication are not meant to be totally comprehensive, but will provide information on techniques and known problem areas.

1. Instrument Drive

The most difficult problems in this assembly will arise from the fabrication of the screw shaft and drive core, pieces I-2 and I-3-2. The square threads specified require that each shaft and core be lathe turned since commercial taps and dies are not available. The internal threads will be the most difficult, requiring the construction of a special boring bar. The large tolerances are required to allow the passage of sea debris through the threads. If the turning of these threads is not feasible a standard Acme screw thread of 2 1/2 x 3 in. - 2G should be used. The die should be adjustable in order to meet the tolerances on the shaft. The use of these threads will increase the insertion time by a factor of 1.5. The power required to insert the instrument would be similarly reduced and will not significantly increase the amp-min. required. The thread tolerances may lead to binding during operation, but only testing will provide an answer.

After assembly, the screw shaft should have little (< 0.10 in.) play longitudinally. Additional washers of Delrin may be placed between the bearing plate and the thrust bearing to accomplish this. After this has been accomplished on a drive, the drive should be disassembled and two thin (0.20 in. thick) foam rubber washers should be placed between the thrust bearings and the bearing plate. Upon reassembly the foam rubber will preload the thrust bearings and damp any shaft vibrations.

The syntactic foam should be cured in place in the I-beam. Prior to filling, threaded aluminum studs can be welded perpendicular to the web of the beam and used, after curing, to hold the flotation in place. Four 1/2 in. diameter studs per side will suffice.

2. Main Frame

The main frame consists of welded pipe and plate and should present no difficulty in fabrication. The cutting pattern, Fig. II-3-4, is provided for piece II-3-4.

The syntactic foam should be cured in paper cylinders slightly less than 7.5 in. in diameter and in various lengths. The cured cylinders can then be placed in the prepared pipes prior to welding end closures. Small blocks and pieces of syntactic foam should be used to fill in around curved end surfaces. The pipes, prior to filling with flotation, should be ported by drilling two 3/8 in. holes per foot length near the top and bottom. This will allow for internal pressure compensation.

3. Coring Cylinder and Ballast Tank

This assembly plus the main frame comprise the majority of the fabrication required for the platform. The coring cylinder and ballast tank are simply designed and will require considerable fabrication time only due to the number of pieces to be assembled. The core guides and mid and end sections must be orthogonal to ensure proper alignment of the core barrels.

The upper cylinder bearing and drive, piece III-1-3, must be isolated electrically from the steel drive sprocket. Plastic electrical tape can be used where Delrin inserts are not provided.

The Delrin bearings can be made up of smaller pieces and bonded to the aluminum by using a cyanoacrylate adhesive, such as Loctite Corporation Super Bonder #6 or an epoxy compound such as Devcon "2-Ton". Surfaces should be thoroughly cleaned prior to application of the adhesive.

After total assembly of the main frame, coring cylinder, and ballast tank, the vertical play should be less than 1/4 in. In air this measurement is the clearance between the upper bearings; in water it is the clearance between the lower bearings. Various thicknesses of the Delrin bearings may be used to meet the requirement and should be determined prior to bonding the bearings to the aluminum.

The core barrel, Fig. II-3-1, requires considerable milling in order to allow for the piano hinge. The hinges can be welded to one half of the core barrel and cold formed to fit the contour of the pipe prior to welding and forming

to the other half. Prior to use of the core barrel, the edges of the halves are sealed with a silicone sealant and the outside of the barrel is uniformly covered with a silicone base grease. The halves can also be sealed by bonding thin strips of foam rubber to the edges.

The bearing portion of the core carrier, piece III-5-1, Fig. III-6-1, should be uniformly covered with a silicone base grease to isolate the steel bearing from the aluminum carrier.

After fabrication of the coring cylinder, Fig. III-1, the flotation can be molded into it. The molding process should be done in layers not to exceed 5 in. in thickness. Thin, flexible plywood (3/16 in.) can be banded to the outside to form the outer wall of the mold. The inner wall, 24.5 in. diameter, can be formed from plywood using additional wooden supports. Stepping this form from the base to the top will fill the void between the ballast tank and the outer edge of the coring cylinder with flotation.

4. Motor Boxes

The motor boxes consist of commercial components housed and framed in a welded 6061-T6 aluminum structure (Fig. IV-1,2). Care must be taken during the assembly of the aluminum boxes and internal frames to ensure accurate alignment of shafts.

After the box fabrication is completed the motor and bearings are loosely attached to the internal frames. The polished shafts and gear sets are assembled starting with the

first stage and proceeding through the remaining stages. Once the gears and bearings are located on a shaft, the set screws are tightened. After the entire gear train is assembled, the bearings can be positioned, to provide for zero backlash in each mesh, and tightened. The shaft seal on the output shaft can then be pressed in place. The shaft keyways can be cut over the entire length of each shaft except the output shaft. The output shaft must be left solid in the area of the shaft seal.

A silicone sealant should be used to seal the cover to the box prior to filling with oil.

The specified bearings have no seals. Should an appropriate, sealed, substitute be used, the seal must be perforated to allow for pressure compensation.

Syntactic foam, in the form of small one inch or larger cubes, is placed in the lower portion of each motor box. The void is to be totally filled with flotation. Plastic ice cube trays will make adequate moulds.

Piece IV-1-9 is used to offset the motor box of the instrument drive being used for coring. The main frame interferes with the normal motor box attachment.

B. ASSEMBLY

Persons associated with the fabrication of the platform will be able to assemble it without further assistance. The following is a short summary of the assembly steps.

1. Lay out and assemble lower main frame.
2. Attach hold down plates.
3. Torque all installed bolts to specifications.
4. Place ballast tank in center of the frame.
5. Lower coring cylinder over ballast tank.
6. Attach main frame uprights and top plate and torque frame bolts to specifications.
7. Center ballast tank, attach frame clamp and torque to specifications.
8. Assemble instrument drives and motor boxes and torque to specifications.
9. Install instrument drive for coring and torque to specifications.
10. Attach battery and control boxes.
11. Attach air flasks to uprights and assemble air piping, regulators and valves.
12. Position instrument drives around main frame and torque to specifications.
13. Attach coring cylinder motor box and drive.
14. Install all limit switches and instruments.
15. Install hold down pumps.
16. Install all electrical cables and cable tubing.
17. Fill compensated systems with oil.
18. Charge air flasks and batteries.
19. With batteries on float, connect umbilical to submersible.
20. Set regulators and check out operation of all electrical systems.
21. Rotate coring cylinder to load cores.
22. Disconnect umbilical and charging systems.
23. Attach standard 4 leg lifting harness.

The platform, with the above completed, is ready for submerged operations.

VI. CONCLUSIONS AND ALTERATIONS

The platform as designed is capable of performing all tasks required for the testing of deep ocean sediments. It can be easily fabricated from readily available materials with normal machine shop capabilities. Once fabricated, the platform can be assembled and disassembled with use of hand tools and a hoist.

The platform is capable of operations to 8,000 ft and is compatible with all forms of sediment testing instruments. The versatility of the design is not limited to the testing of sediments, but can be easily adapted to any thalassic operation requiring a bottom based platform.

The design represents a practical limit in the use of air as a ballasting medium due to the weight of the air at operating depth as described in Section IV-D. The pressures and volumes of air that would be required to operate much deeper than 8,000 ft would generate enormous engineering problems and requirements for materials not yet developed. The literal limiting depth is of course 50,300 ft, where the density of air would equal the density of sea water. For purposes of comparison, the density of air at 8,000 ft is approximately 2/7 that of sea water.

An enclosed ballasting system would be necessary for operations greater than 8,000 ft. The system would consist of a ballast tank, air flasks, and a blow valve similar to

the installed air system. There would be a compressor in place of the vent valve. The compressor would transfer air, at relatively low pressure in ballast tanks, to the air flasks at a higher pressure. If the displaceable volume of the ballast tank is small, implying a very accurate compensation, the differential pressure across the compressor will be small and will require little power to operate. The operation of this system is limited only by the power available since there is no mass transfer from the platform by venting. The cost of the compressor and additional power requirements weighed against the cost of the replaced vent valve and two pressure flasks provided the incentive for the final design.

As a further explanation of the last statement, if the flask pressure is 8,000 psi, and the ballast tank is as designed, the volume of air required would be 15.5 ft^3. This would eliminate the need for two of the air flasks.

A drawback of the compressor operated system, which must be considered, is the rate of ballast changing. The compressor capacity will, by necessity, be small and hence the rate of decreasing buoyancy will be low. A rapid decrease in buoyancy would be necessary in order to stop the ascent of the platform in emergencies. A venting system could be installed for emergency use, but once used, the platform may still remain in an uncontrollable state due to the loss of air weight. With over a mile of water above one, this condition could be considered less than desirable.

The only alteration that is presently known to be desirable under special circumstances is the lengthening of the core barrel penetration from 3 ft to 4 ft. This can be accomplished by lengthening pieces I-2 and I-5 by 12 in. and redesigning piece III-4-1 to allow the instrument block to be raised 12 in. relative to the core barrel lifting bearing. The overall factor of safety, relative to the 500 lb insertion load, is reduced to approximately 2. The critically loaded portion of the device is the upper and lower sleeve bearings of Delrin AF. If Rulon LD, product of Dixon Corp., is substituted for Delrin AF, the overall safety factor will be increased to above 3. The PV limit for Rulon LD is 10,000.

The hold down test plate should be operated in typical ocean sediments to provide an accurate pressure profile across the plate. Once the profile is established, variations in the plate diameter may have to be made. The determination of an adequate hold down force should be made by comparing the load required to insert an instrument into the same sediment (Section IV-13). If they are nearly equal, with a safety factor of 3 applied to the insertion load, the hold down force should be considered as adequate. A vane shear measurement should be made in the sediment at the time of testing the hold down plate.

APPENDIX A

CALCULATIONS

A. DESIGN CALCULATIONS FOR THE INSTRUMENT DRIVE ASSEMBLY

1. Design of the Screw Shaft

It is desired to know the deflection of the screw shaft at the instrument mount block to determine the clearance between the frame and the mount block (Fig. I).

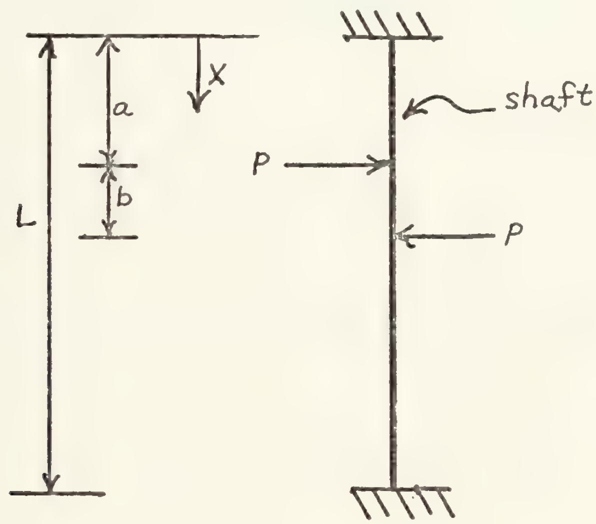


FIG. 1 LOADING OF SHAFT

The shaft was considered to be rigidly supported by the end bearings (Fig. 1). The couple Pb, is the result of the offset loading of the mount block (Fig. I-4) during insertion. (For withdrawal, P is negative.) For design considerations, the offset is considered to be 7 in., measured from the centerline of the screw shaft to the line of load application. This allows for the load application to be centered 2 in. to the left of the mount block face (Fig. I).

The design insertion and withdrawal load is 1500 lb, which includes a safety factor of 3 over the expected load of 500 lb. This yields a couple of 10,500 in-lb on the shaft. The dimension "b" is the length of the threaded portion of the mount block (Fig. I-3-2, b = 7.25 in.). Since Pb must equal the applied couple, it follows that P is 1448.3 lb.

a. Solving the Statically Indeterminant Beam Problem

The general equation for the shear in the beam was written and directly integrated to yield the following [5, p. 400-403]:

$$EI\,\frac{d^3v}{dx^3} = V = P\langle x-a\rangle^0 - P\langle x-a-b\rangle^0 + R \qquad (1)$$

$$EI\,\frac{d^2v}{dx^2} = M' = P\langle x-a\rangle^1 - P\langle x-a-b\rangle^1 + Rx + M \qquad (2)$$

$$EI\,\frac{dv}{dx} = EI\theta = \frac{P}{2}\langle x-a\rangle^2 - \frac{P}{2}\langle x-a-b\rangle^2 + \frac{Rx^2}{2} + Mx + A \qquad (3)$$

$$EI\,v = \frac{P}{6}\langle x-a\rangle^3 - \frac{P}{6}\langle x-a-b\rangle^3 + \frac{Rx^3}{6} + \frac{Mx^2}{2} + Ax + B \qquad (4)$$

where R and M are the reaction and the moment, respectively, at the top of the shaft (X = 0.0). A and B are constants of integration. All loading positive when directed to the right of Fig. 1 and all clockwise moments are positive.

Applying the boundary conditions, the slope, θ, and deflection, v, of the shaft are zero at both ends, to Equations (3) and (4) yields the following solutions:

$$A = B = 0.0$$

$$R = \frac{2P\alpha^3}{L^3} - \frac{3P\alpha^2}{L^2} - \frac{2P\beta^3}{L^3} + \frac{3P\beta^2}{L^2} \tag{5}$$

$$M = -\frac{P\alpha^3}{L^2} + \frac{P\alpha^2}{L} + \frac{P\beta^3}{L^2} - \frac{P\beta^2}{L} \tag{6}$$

where;

$$\alpha = L - a$$

$$\beta = L - a - b$$

b. Moments, Deflections, and Reactions

The preceeding equations were evaluated for the mount block positioned at the center, 2/3 of the length from the top, and at the bottom of the shaft. Table 1 exhibits the results for the shaft shown in Fig. I-2. The shaft is 60 in. long and 2 in. in diameter. The material is steel with a modulus of 30×10^6 psi.

Superposition of equations presented by S. Timoshenko [6, p. 186-187] was used to check the moments at the ends of the beam.

The effect of the screw threads in strengthening the shaft was neglected. The maximum positive deflection obtained was 0.02 in., when the slide was approximately 2/3 of the distance from the top of the shaft. Figures I-1,

TABLE 1

TABLE OF DEFLECTIONS, MOMENTS, AND REACTIONS

	CASE I	CASE II	CASE III
a	26.4 in.	36.4 in.	51.5 in.
b	7.25 in.	7.25 in.	7.25 in.
L	60.0 in.	60.0 in.	60.0 in.
P	1448.3 lb	1448.3 lb	1448.3 lb

	x in.	DEFLECTION in.	MOMENT in-lb	REACTION lb
CASE I	0.0		2591	- 261
	26.45	0.0043	-4305	
	33.65	-0.0042	4301	
	60.0		-2582	+ 261
CASE II	0.0		3462	- 232
	36.4	0.018	-4979	
	43.6	0.0074	3838	
	60.0		38.33	+ 232
CASE III	0.0		1460	- 77.0
	51.5	0.0077	-2511	
	58.75	0.00024	7430	
	60.0		7334	+ 77.0

I-4-1 show a minimum clearance of 0.08 in. between the frame and the mount block. The maximum displacement of the tip of an instrument 60 in. long occurs at this point and is approximately 0.09 in. This was considered as an acceptable displacement for the instrument tip during insertion. This is also the maximum displacement of the tip during withdrawal; when the mount block is 1/3 the distance to the top of the screw shaft.

c. Bending Stresses in the Shaft

As seen from Table 1, the maximum bending moment can be approximated by 7500 in-lb. This was used as the design moment for end reactions. The bending stress is found from the following equation:

$$\sigma_b = \frac{My}{I} \tag{7}$$

where;

M = applied moment (in-lb)

y = distance from neutral axis of cross section to outer fibers (in.)

I = area moment of inertia (in^4)

solving;

$$\sigma_b = \frac{(7500)(1.032)}{(0.88913)}$$

$$\sigma_b = 8700 \text{ psi}$$

The compressive stress due to the 1500 lb insertion load is 477.5 psi. Thus the total stresses are, 9178.4 psi compressive, and 8223.4 psi tensile. This provides an additional safety factor of at least 6 for normal steels.

d. Buckling of the Shaft

Considering the extreme case of the shaft being built in at one end and hinged at the other, with the load applied to the hinged end; the following equation provides for the critical buckling load of the shaft [14, p. 89]:

$$P_{cr} = \frac{20.16\ EI}{L^2}$$

where;

E = elastic modulus (psi)

I = area moment of inertia, $0.88913\ in^4$

L = length of the shaft (in.)

solving;

$$P_{cr} = \frac{20.16(30 \times 10^6)(0.88913)}{(60)^2}$$

$$P_{cr} = 149{,}400\ \text{lb}$$

The 1500 lb load will not buckle the shaft and it is safe to assume that the actual offset loading will not contribute significantly enough to buckle the shaft.

2. Design of the Upper Bearing Plate

The upper bearing plate (Fig. I-1) is bolted to the instrument drive frame (Fig. I-6) to facilitate servicing the screw shaft and the mount block. The plate is loaded by the 1500 lb design insertion load and the moment due to the bending of the screw shaft by the mount block. This applied moment is maximum when the mount block is at the top of the shaft and in the process of insertion under design load. The maximum moment for design purposes, as previously stated, is 7500 in-lb.

a. Deflection of the Bearing Plate

Superposition of equations presented by R. Roark [7, p. 104-106] for tip loaded cantilever beams was used to solve for the maximum tip deflection under design loading. The plate is assumed to be clamped at the drive frame.

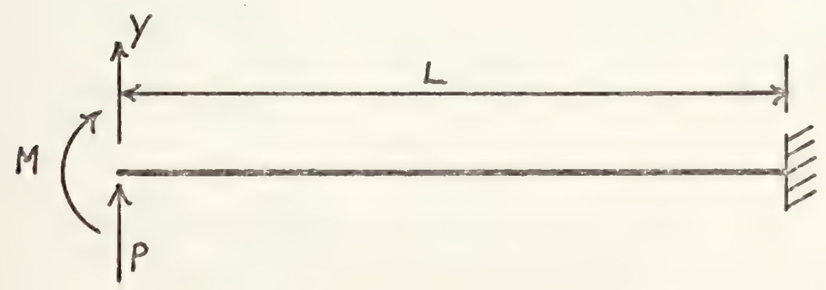


FIG. 2 LOADING OF THE UPPER BEARING PLATE

The equation for tip deflection is:

$$Y = \frac{P}{6EI}(2L^3) + \frac{M}{2EI}(L^2) \tag{8}$$

where;

P = design load (lb)

M = design moment (in-lb)

I = area moment of inertia , $8/3\ in^4$

E = elastic modulus , 10×10^6 psi (aluminum)

L = length of cantilever , 5.5 in.

solving;

$$Y = \frac{(1500)(2)(5.5)^3(3)}{6(8)(10 \times 10^6)} + \frac{(7500)(5.5)^2(3)}{2(8)(10 \times 10^6)}$$

$$Y = 0.0074 \text{ in.}$$

This is considered to be well within the tolerances of the design.

b. Maximum Stress in Bearing Plate

The maximum stress due to bending occurs at the root of the cantilever and is found from Equation (7).

$$\sigma_b = \frac{M'Y}{I}$$

where;

M' = moment at root, 1500(5.5) + 7500 in-lb

I = area moment of inertia, 8/3 in^4

Y = distance from outer fiber to neutral axis, 1.0 in.

solving;

$$\sigma_b = \frac{[7500 + 1500(5.5)](1.0)}{(8/3)} = 5910 \text{ psi}$$

Since the safety factor of 3 is incorporated in the design load of 1500 lb, this maximum stress is within the limits of available aluminum plates. The recommended material is 6061-T6 aluminum, with a yield stress of 40,000 psi.

c. Fasteners for the Bearing Plate

The bearing plate is fastened to the drive frame by four bolts as shown in Fig. I-1. The loading of the plate is shown in the following figure.

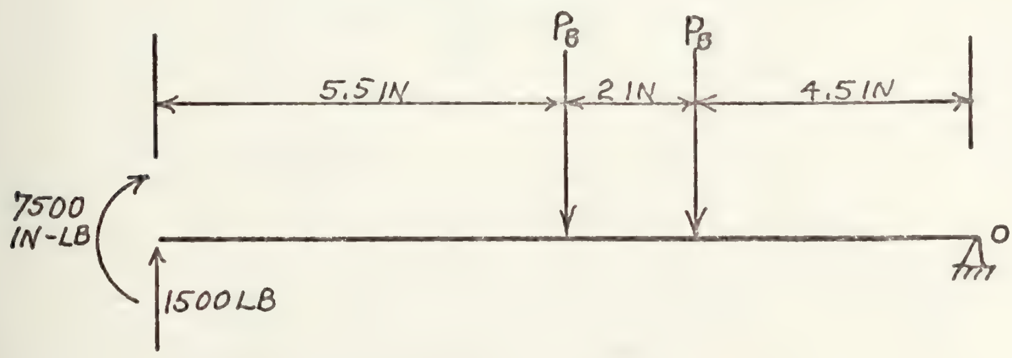


FIG. 3 LOADING OF BEARING PLATE FASTENERS

The required bolt preload is solved for by summing moments about point "o" in Fig. 3 and solving for the required load for equilibrium. This yields a required load of 1159.1 lb for each bolt. The constraints of the structure are such that a 9/16 in. diameter bolt would be the largest acceptable. The following equation is provided for calculation of the allowable loads on a bolt: [8, p. 159]:

$$Fe = \frac{\sigma_y (As)^{\frac{3}{2}}}{6} \tag{9}$$

where;

Fe = bolt load (lb)

σ_y = yield strength of material (psi)

As = stress area of bolt (in^2)

The stress area of a 9/16 in. bolt is 0.1820 in^2 [8, p. 588]. The bolt material is K-Monel with a yield strength of 111 x 10^3 psi [9]. Solving;

$$Fe = \frac{(111 \times 10^3)(0.182)^{3/2}}{6}$$

$$Fe = 1436 \text{ lb}$$

When torqued properly the 9/16 in. bolts will keep the bearing plate from lifting.

The following equation is provided for the required torque [8, p. 159].

$$T = C\ D\ Fe \tag{10}$$

where;

T = tightening torque (in-lb)

C = empirical coefficient

D = diameter of the bolt (in.)

Fe = required load (lb)

Values for C as 0.20 for an "as received condition" and 0.15 for a "lubricated condition" are given by V. Faires [8, p. 159]. Due to typical fabrication practice, the value of C equal to 0.20 was selected. Solving;

$$T = 0.20(\frac{9}{16})(1436) = 161.6 \text{ in-lb}$$

3. Design of the Lower Bearing Plate

The lower bearing plate is fully welded to the drive frame and was designed by the same procedure as for the upper bearing plate.

a. Deflection of the Bearing Plate

The solution is the same as Section A-2-a, App. A, using Equation (8) and the loading of Fig. 4.

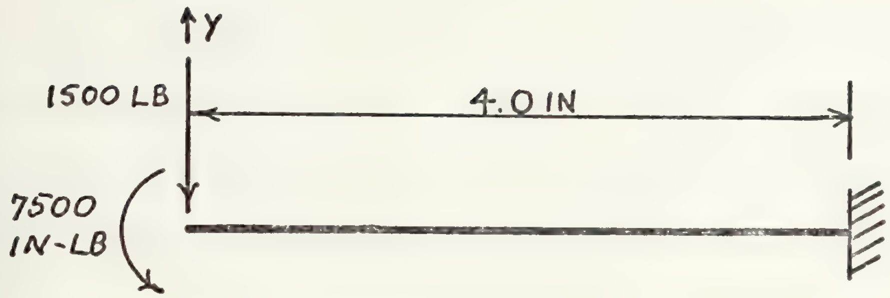


FIG. 4 LOADING OF THE LOWER BEARING PLATE

Solving Equation (8);

$$Y = \frac{-1500(2)(4.0)^3(3)}{6(8)(10 \times 10^6)} + \frac{-7500(4.0)^2(3)}{2(8)(10 \times 10^6)}$$

$$Y = -0.0035 \text{ in.}$$

This deflection is well within the limits of the design.

b. Maximum Stress in the Bearing Plate

The solution is the same as presented in Section A-2-b, App. A, using Equation (7). Solving Equation (7);

$$\sigma_b = \frac{[7500 + 1500(4.0)](1.0)}{(8/3)}$$

$$\sigma_b = 5063 \text{ psi}$$

This allows for an additional factor of safety of 7.9 due to the 40,000 psi yield strength of 6061-T6 aluminum.

c. Strength of the Weld

The loading of the weld between the bearing plate and the drive frame is shown in Fig. 5. The width of the weld bead is assumed to be 1/2 in. and the I-beam is to be fully welded to the bearing plate.

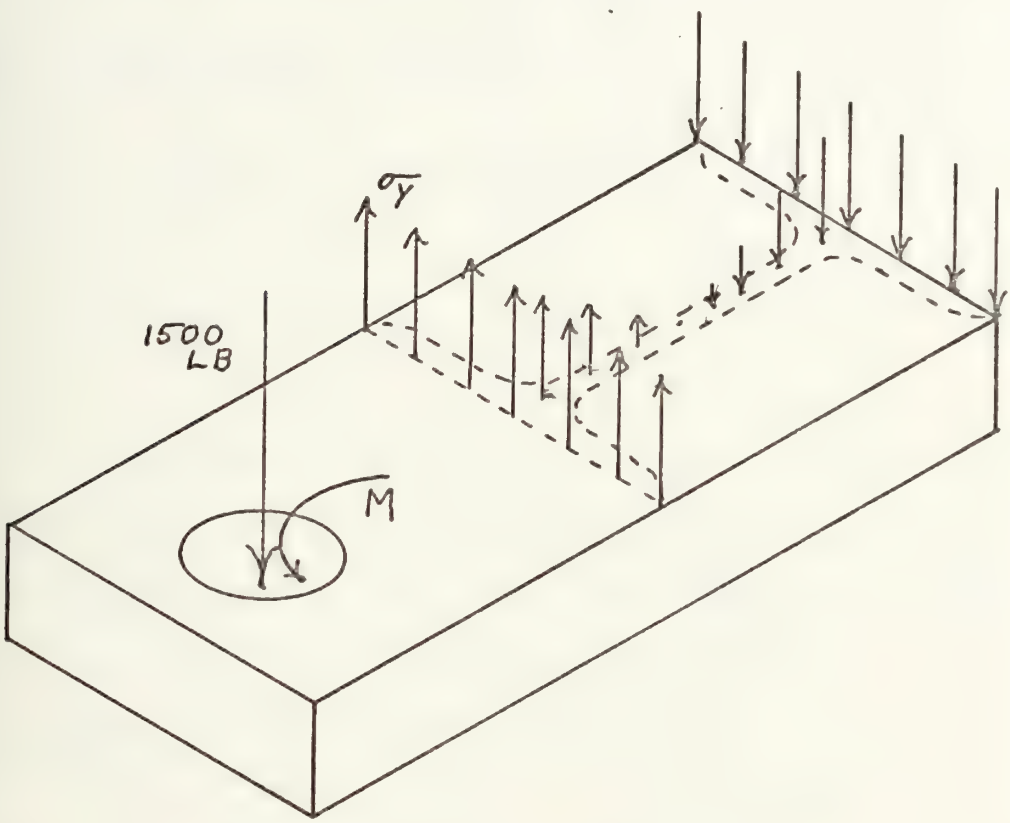


FIG. 5 LOADING OF THE WELD

In order to determine the stress in the weld, the uniform tensile stress due to the design load was added to Equation (7) and evaluated at the outer fibers of the weld.

$$\sigma_y = \frac{1500\ \text{LB}}{A} + \frac{My}{I} \tag{11}$$

where;

I = area moment of inertia , 85.90 in^4

y = outer fiber , 4.5 in.

A = cross sectional area of weld , 6.38 in^2

σ_y = yield stress of the weld , 43×10^3 psi [10]

solving for the unknown moment "M"

$$M = \frac{85.90}{4.5} \left(43 \times 10^3 - \frac{1500}{6.38}\right)$$

$$M = 8.16 \times 10^5 \text{ in-lb}$$

The moment allowed by the weld exceeds the maximum moment that the plate or I-beam will support in bending. The excessive strength of this weld is common to all welds in the system and will not be demonstrated further.

4. Design of the Instrument Drive Frame

The drive frame is the main strength member of the instrument drive assembly and is the structural link between the instrument drive and the platform. Since deflection of the member in bending was the main criterion for design, an I-beam was chosen as the member shape. The constraints of the drive system require the minimum width of the beam to be about 4 in. An American Standard Aluminum I-beam, 4 in. wide by 8 in. deep, was selected. For design purposes the drive frame was considered to be rigidly fixed at a position coincident with the top of the main frame member of the platform.

The design loading of the I-beam is shown in Fig. 6.

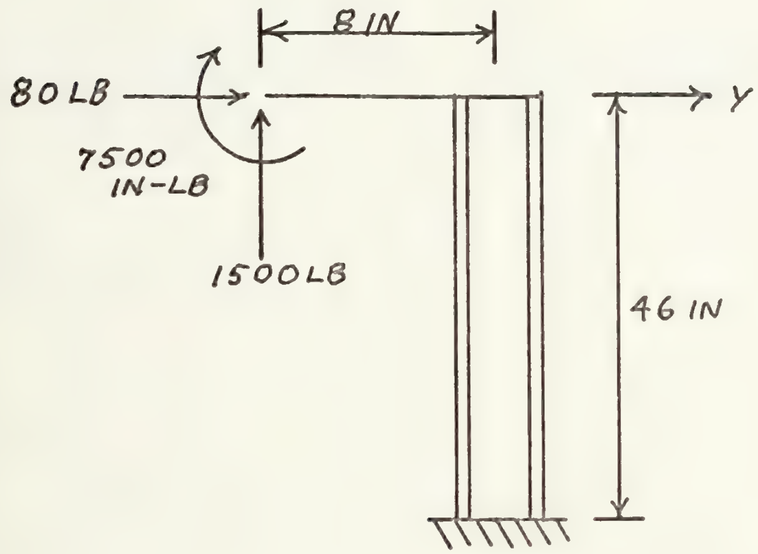


FIG. 6 LOADING OF THE DRIVE FRAME

The loads are all the result of the offset 1500 lb design insertion load reacting through the screw shaft and upper bearing plate.

a. Deflection of the Drive Frame

Applying Equation (8), as in Section A-2-a, App. A, to the loading of the drive frame, the following results were obtained: solving Equation (8) where;

I = area moment of inertia , 57.55 in^4

E = elastic modulus , 10 x 10^6 psi, aluminum

L = length of beam , 46 in.

P = tip load, 80 lb

M = tip moment, 1500 (8) + 7500 in-lb

$$y = \frac{80(2)(46)^3}{6(57.55)(10 \times 10^6)} + \frac{[7500 + 1500(8)](46)^2}{2(57.55)(10 \times 10^6)}$$

y = 0.0404 in.

This deflection results in a tip displacement of a fully inserted instrument of 0.053 in. This added to the tip displacement of 0.064 in. caused by the bending of the screw shaft yields a total displacement of 0.117 in. from the vertical. The maximum total displacement of the instrument tip occurs when the mount block is approximately 2/3 the distance from the top of the screw shaft; or, the instrument has been inserted 2/3 of its distance of travel. This displacement is 0.130 in. A displacement of this magnitude was considered acceptable for the instrument tip during insertion.

During withdrawal the lower portion of the I-beam would be loaded similar to Fig. 6 and the deflection of the beam would be 0.0005 in. This deflection and the associated stresses are clearly negligible for design purposes.

b. Maximum Stress in the Drive Frame

Equation (11) is applied to the loading in Fig. 6 in order to calculate the bending stress at the root of the cantilever. Solving Equation (11) where;

M = sum of the moments acting at the root of the beam = 80(46) + 1500(8) + 7500 in-lb

y = outer fiber distance , 4 in.

P = axial load , 1500 lb

A = cross sectional area , 5.4 in^2

$$\sigma_{max} = \frac{[80(46) + 1500(8) + 7500](4)}{57.55} + \frac{1500}{5.4}$$

$$\sigma_{max} = 1890 \text{ psi}$$

This stress is well within the limits of the yield strength of a 6061-T6 aluminum.

5. Design of the Mounting Brackets

The mounting brackets (Fig. I, I-5) hold the instrument drive assembly to the main frame of the platform. The loading of the mount bracket is shown in Fig. (7).

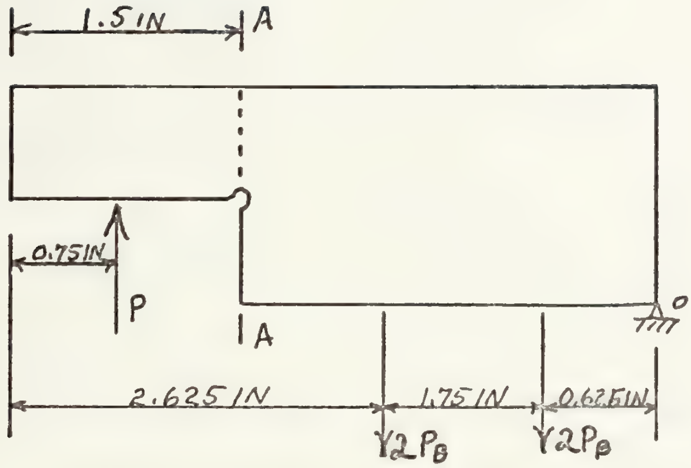


FIG. 7 LOADING OF THE MOUNT BRACKET

The load, P, is the result of the only external force on the drive assembly, the design load of 1500 lb. The determination of the load, P, is shown in Fig. 8.

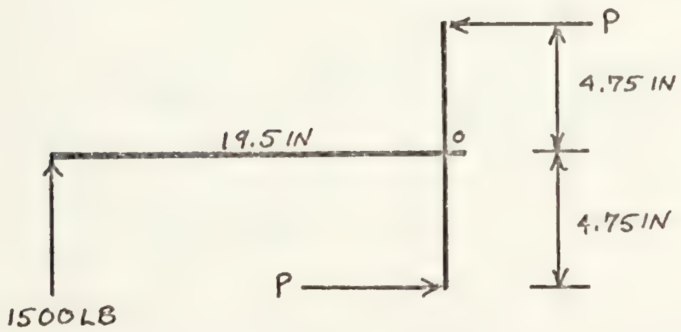


FIG. 8 LOAD TRANSFER TO THE MAIN FRAME

Summing moments about the point "o" yields the following results.

$$P = \frac{1500(19.5)}{2(4.75)}$$

$$P = 3080 \text{ lb}$$

P_b represents the force exerted by the preloaded bolts fastening the mounting bracket to the drive frame. There are four such bolts for each mounting bracket.

a. Maximum Stress in the Mounting Bracket

The maximum stress occurs at Section A-A (Fig. 7) of the mounting bracket. Applying Equation (7) the following results are obtained. Solving Equation (7) where;

M = moment applied

I = area moment of inertia , 0.2233 in^4

y = outer fiber distance , 0.4375 in.

K = stress concentration factor , 1.73 [13, p. 139]

$$\sigma_{max} = \frac{My(K)}{I} = \frac{3080(0.75)(0.4375)(1.73)}{0.2233}$$

$$\sigma_{max} = 7830 \text{ psi}$$

This stress is well within the yield strength of 6061-T6 aluminum and provides an added safety factor of approximately 5.1.

b. Fasteners for the Mounting Bracket

Summing the moments about point "o" in Fig. 7 yields the required bolt preload for equilibrium of the mounting bracket.

$$P_B = \frac{3080(4.25)}{4(1.5)}$$

$$P_B = 2180 \text{ lb}$$

The bolts selected were 3/4 in. in diameter. Equation (9) was used to solve for the allowable yield stress of the bolt material. The stress area of a 3/4 in. bolt is 0.334 in^2 [8, p. 588].

$$\sigma_y = \frac{6\ P_B}{(A_s)^{3/2}} \tag{12}$$

where;

P_B = required preload (lb)

A_s = stress area (in^2)

solving;

$$\sigma_y = \frac{6(2180)}{(0.334)^{3/2}}$$

$$\sigma_y = 67{,}800 \text{ psi}$$

This is well below the yield stress of K-Monel, (111×10^6 psi), which was the material chosen for the bolts.

The torque required to preload the bolts was found from Equation (10). The coefficient for the as received condition, 0.20, was used. Solving Equation (10) where;

D = nominal diameter of the bolt , 0.75 in.

P_B = required preload , 2180 lb

C = 0.20

$$T = 0.20(0.75)(2180)$$

$$T = 327.1 \text{ in-lb}$$

6. <u>Design of the Thrust Bearings</u>

The thrust bearings, located at the ends of the screw shaft, carry the axial design loading of the screw shaft due to insertion or withdrawal of an instrument. The thrust bearing will also support a major portion of the moment required to hold the end of the shaft rigidly at zero slope. This moment, as stated before, has a maximum value of 7500 in-lb. For design purposes, the thrust bearing will be considered as supporting this moment.

An open race, ball type, thrust bearing was chosen for the design. The open design would allow for lubrication of the bearing by sea water.

a. Loading of the Bearing

Considering a bearing with an inside diameter of 2.75 in. and an outside diameter of 4.0 in. for the design, the resulting race diameter of 3.41 in. yields a total moment arm of 1.7028 in. to resist the design moment of 7500 in-lb. Assuming a linear force distribution along the bearing race, Fig. 9 represents the loading of the bearing due to the design moment.

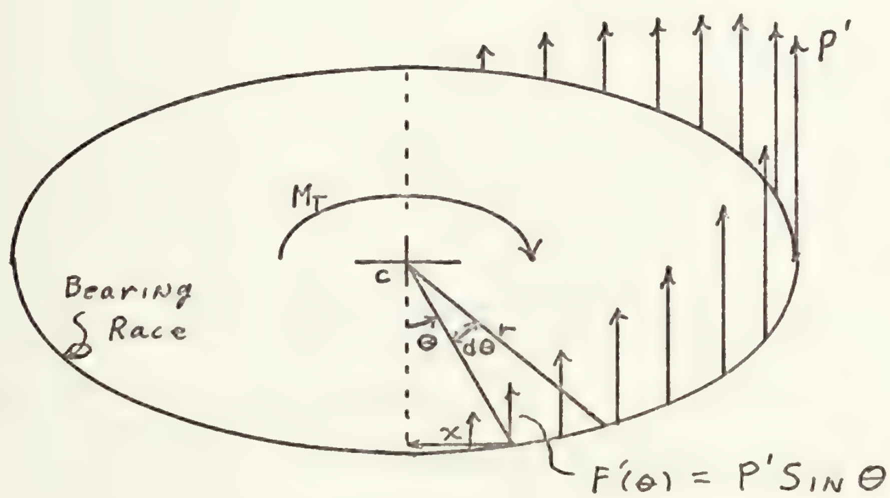


FIG. 9 LOADING ON THE THRUST BEARING DUE TO THE DESIGN MOMENT

Summing moments about the center of the bearing race, point C, the following equation results.

$$M_T = 2r^2\rho' \int_0^{\pi/2} \sin^2\theta \, d\theta \qquad (13)$$

where;

M_T = design moment (in-lb)

r = radius of the bearing race , 1.375 in.

ρ' = maximum thrust load/inch of bearing race (lb/in)

Solving for thrust load/inch;

$$\rho' = \frac{M_T}{2r^2 \int_0^{\pi/2} \mathrm{Sin}^2\theta \, d\theta}$$

$$\rho' = \frac{2M_T}{r^2 \pi}$$

$$\rho' = 2525.4 \text{ lb/in}$$

A bearing with stock no. 914, produced by SKF Industries, Inc. of Philadelphia, Pa. was selected. The dynamic rated load is 13,200 lb distributed over 20 balls. This allows for a rated load of 660 lb per ball.

In order to determine the design load per ball, a ball was assumed to be positioned at theta equalling 90 degrees (Fig. 9). The load supported by this ball can be determined from the following equation:

$$F = 2a\rho' \int_{\theta_1}^{\theta_2} \mathrm{Sin}\theta \, d\theta$$

where;

F = the load on the ball (lb)

a = inches of ball race circumference/radian, 1.70275 in/rad

θ_1 = angle in degrees, to the center of the space between the subject ball and the adjacent ball, 81°

θ_2 = angle in degrees, to the center of the subject ball, 90°

ρ' = maximum thrust bearing load/inch of race, 2803.99 lb/in

solving;

$$F = 2\rho' a \text{ Cos } \theta_1$$

$$F = 2(2525.4)(1.70275) \text{ Cos } 81°$$

$$F = 1345 \text{ lb/ball}$$

Adding the load per ball due to the axial load of 1500 lb, the maximum total load per ball was found to be 1420 lb. This load which includes the safety factor of 3, applied to the expected insertion load, exceeds the allowable load of 660 lb per ball. Using the expected insertion load of 500 lb, the total load per ball reduces to 523 lb. When compared to the allowable load, the safety factor is 1.26. The manufacturer [12, p. 154] provides that this bearing,

operating with a safety factor of 1.26, will have a minimum life of 240 hours at a design speed of 120 RPM. Considering that an operational cycle of the instrument drive requires the loading of each bearing for one minute, the minimum life of each bearing is 14,400 cycles. This is well within the design criterion of the system.

The manufacturer states that permanent deformation of the balls will occur if the load exceeds 1775 lb per ball. This is greater than the design load per ball for a safety factor of 3. Since the design load is not expected and if it were to occur it would be only momentary, this bearing is expected to function properly within the system. The ease of replacement and relatively low cost also enhance the acceptance of this marginal design.

7. Torque Required to Rotate the Screw Shaft

The torque required to rotate the shaft is the sum of the friction torques in the thrust and sleeve bearings and the friction in the power threads.

a. Power Thread Torque

The following equation provides the torque required to rotate the power thread under load (Fig. I-3) [8, p. 249].

$$T_p = \frac{P\ D}{2}\left(\frac{\operatorname{Tan} \lambda + f}{1 - f \operatorname{Tan} \lambda}\right) \qquad (14)$$

where;

T_p = torque to rotate (in-lb)

D = mean diameter of the screw , 2.25 in.

$\tan \lambda$ = lead of the screw/mean diameter , 0.0707

f = coefficient of friction , 0.125, [8, p. 250]

P = axial load on the screw (lb)

solving;

$$T_p = \frac{1500(2.25)}{2} \left(\frac{0.0707 + 0.125}{1 - 0.125(0.0707)}\right)$$

$$T_p = 333 \text{ in-lb}$$

b. Thrust Bearing Torque

The coefficient of friction for the thrust bearings is 0.0013, measured at the radius of the bore of the bearing [12, p. 16]. The following equation applies:

$$T_t = R\ P\ (0.0013) \tag{15}$$

where;

T_t = torque to rotate (in-lb)

R = radius of the bore , 1.072 in.

P = thrust load , 11,500 lb

solving;

$$T_t = 1.072(11{,}500)(0.0013)$$

$$T_t = 16.0 \text{ in-lb}$$

c. Sleeve Bearing Load and Total Torque

The maximum sleeve bearing (Fig. I-1) torque is due to the sum of the load due to bending of the screw and the tension in the chain drive during operation. This load is subsequently dependent upon the total torque required to rotate the screw shaft and the radius of the chain sprocket. The radius of the sprocket is 3.3 in. The coefficient of friction of Delrin AF is approximately 0.10 as provided by the DuPont Company. The following equations were solved to yield the sleeve bearing load and the total torque required to rotate the shaft.

$$P = P_s + \frac{T}{3.3} \tag{16}$$

$$T = R_s\, f\, P + T_p + T_t \tag{17}$$

where;

T = total torque to rotate the shaft (in-lb)

P = total load on the bearing (lb)

P_s = load on the bearing due to bending of the screw shaft (Table I)(lb)

R_s = radius of the shaft (in.)

f = coefficient of friction , 0.10

T_p & T_t = torques previously calculated (in-lb)

solving;

$$P = 260 + \frac{T}{3.3}$$

$$T = 1.032(0.10)(P) + 333 + 16.0$$

$$P = 376 \text{ lb}$$

$$T = 388 \text{ in-lb}$$

d. Power Required to Rotate the Screw Shaft

The shaft rotates at 120 RPM. The following equation provides the horsepower to rotate the shaft:

$$HP = \frac{\pi N T}{6(33{,}000)}$$

where;

N = RPM

T = torque (in-lb)

solving;

$$HP = \frac{\pi(120)(388)}{6(33,000)}$$

$$HP = 0.739 \text{ hp}$$

8. Upper and Lower Sleeve Bearing Design

Delrin AF, produced by DuPont Co., Wilmington, De. was selected as the material for the bearings (Fig. I-1). Its dielectric properties make it an excellent insulator against corrosion between the steel shaft and the aluminum bearing plate. Delrin is relatively inexpensive and is easily machined.

a. Load Capacity

The manufacturer provides a "PV" limit for unlubricated bearings of 5800 for a bearing velocity of 62 ft/min. The "PV" number is the product of the bearing load in pounds per square inch of projected shaft area and the surface velocity of the shaft in feet per minute. The design load is 376 (Section A-7-c, APP. A). The projected shaft area is 4 in^2 and the surface velocity of the shaft is 62.8 ft/min. This yields a "PV" value of 5915, which exceeds the manufacturer's specification for unlubricated bearings by 2%. The manufacturer states that if the bearing is lubricated the "PV" limit will increase, but provides no quantitative value of lubrication. Since the design provides an overall safety

factor of 2.94 it is expected to operate satisfactorily within the system. The "PV" value for the expected insertion load of 500 lb is 1971.

B. DESIGN CALCULATIONS FOR THE MAIN FRAME

Since the actual loading of the frame is small compared to the capacity of the frame to carry the load, the following calculations will demonstrate the maximum allowable loading. Figure II exhibits the main frame and its various components. Figures II-1 through II-4-1 exhibit the actual construction of each component.

1. Maximum Torsion of the Lower Frame Members

The lower frame members are the main loading areas of the frame (Fig. II-3,4). The instrument drive assemblies are designed to be mounted anywhere on the lower frame members. The 1500 lb design insertion load, offset by 19.5 in. from the center of the frame member, provides the largest expected torque within the frame. This torque is 29,250 in-lb (Fig. 8). A typical cross section of a lower frame member is shown in Fig. (10).

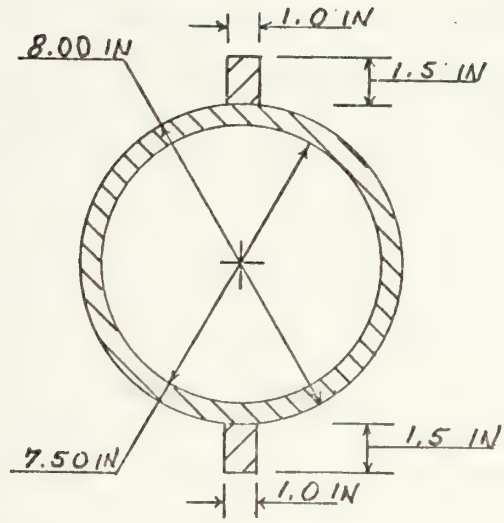


FIG. 10 CROSS-SECTION OF TYPICAL LOWER FRAME MEMBER

There are many methods capable of resolving the maximum torsion the member will support. The finite element method would provide the most accurate results and would include the influence of the stress concentrations where the rectangular bar is welded to the pipe. A solution of this intricacy would, as will be demonstrated later, require an inordinate expenditure of resources. S. Timoshenko [16, p. 237] outlines the use of membrane analogy as a method of stress analysis in torsional members of non-standard cross section. This method would also include the effects of the stress concentrations, but, again, would not be desirable since the required apparatus is not available. The method used here, although approximate, proves to be sufficiently accurate for our purpose. It divides the cross section into various standard cross sections, for which the required equations are

known. A selected portion of the cross section is subjected to torsion and the remaining portions are subjected to a combination of bending, in two orthogonal directions, and torsion, in order to maintain rigid body continuity. This method assumes that the cross section will remain unchanged by the torsion and neglects the effects of the stress concentrations.

The subject cross section, Fig. 10, can be divided into three standard cross sections; a hollow cylinder and two rectangular bars. The cylinder was subjected to torsion and the bars were bent and torqued to comply with rigid body motion. The radial load, relative to the centerline of the cylinder, required to bend the bar into the rigid body position after torsion of the cylinder, is less than 10 lb and will be neglected in the following calculations.

The following equations provide the torsion of a hollow cylinder [6, p. 288].

$$\theta = \frac{32\ M_c}{\pi(D^4 - d^4)\ G} \tag{18}$$

$$\tau_{max} = \frac{16\ M_c}{\pi\ D^3(1 - \frac{d^4}{D^4})} \tag{19}$$

where;

θ = angle of twist per unit length (rad/in)

τ_{max} = maximum shear stress in the section (psi)

M_c = torsional moment (in-lb)

D = outside diameter, 8.00 in.

d = inside diameter, 7.50 in.

G = shear modulus of elasticity, 3.75×10^6 psi

Equations (18) and (19) were solved for the allowable moment and angle of twist at a yield strength of 30×10^3 psi in shear for 6061-T6 aluminum.

Solving Equation (19) for M_c;

$$M_c = \frac{30{,}000(\pi)(8.00)^3\left(1 - \frac{(7.5)^4}{(8.0)^4}\right)}{16}$$

$$M_c = 686{,}200 \text{ in-lb}$$

Solving Equation (18) for theta;

$$\theta = \frac{32(686{,}195)}{\pi((8.0)^4 - (7.5)^4)(3.75 \times 10^6)}$$

$$\theta = 0.002 \text{ rad/in}$$

The following equations provide the torsion of a bar with a rectangular cross section [6, p. 289].

$$\theta = \frac{M_b}{\beta\ b\ c^3\ G} \tag{20}$$

$$\tau_{max} = \frac{M_b}{\alpha\ b\ c^2} \qquad (21)$$

where;

θ & τ_{max} = as defined before

M_b = torsional moment (in-lb)

b = the long side of the section, 1.50 in.

c = the short side of the section, 1.00 in.

G = shear modulus, 3.75×10^6 psi

α & β = constants [6, p. 290]

$\alpha = 0.231$
$\beta = 0.196$

Equations (20) and (21) were solved for the allowable moment and angle of twist at a yield strength of 30×10^3 psi in shear for 6061-T6 aluminum.

Solving Equation (21) for M_b;

$$M_b = 30{,}000(0.231)(1.5)(1.0)^2$$

$$M_b = 10{,}395 \text{ in-lb}$$

Solving Equation (20) for theta;

$$\theta = \frac{10{,}395}{0.196(1.5)(1.0)^3(3.75 \times 10^6)}$$

$$\theta = 0.0094 \text{ rad/in}$$

Since the section twists as a rigid body, the angle of twist of the cylinder and the bars must be the same. Equating Equations (18) and (20) yields the following relationship between M_c and M_b for this specific cross section.

$$M_b = \frac{32\ \beta\ b\ c^3}{\pi(D^4 - d^4)} M_c$$

Solving;

$$M_b = \frac{32(0.196)(1.5)(1.0)^3}{\pi((8.0)^4 - (7.5)^4)} M_c$$

$$M_b = \frac{1}{311} M_c \qquad (22)$$

The cylinder yields at the lowest angle of twist and thus limits the angle of twist of the composite section to 0.002 rad/in. The total moment supported by the section, M_s, exclusive of the torque required to bend the bars, can be represented as a function of the limiting moment, M_c, as follows.

$$M_s = M_c + 2\ M_b$$

Applying Equation (22);

$$M_s = (1 + \frac{2}{311}) M_c \tag{23}$$

Solving for M_s with M_c previously calculated;

$$M_s = (1 + \frac{2}{311})(686,200)$$

$$M_s = 690,600 \text{ in-lb}$$

If the torsionally loaded frame member is assumed to be rigidly fixed at both ends and it is loaded symetrically, the maximum allowable moment would be 1.38×10^6 in-lb. With a safety factor of 3, the maximum allowable moment would be 4.6×10^5 in-lb. This is in excess of any expected load by a factor of approximately 16.

The tangential load on the rectangular bar, required for rigid body continuity is 24 lb/in for a member 15 in. long, with a torsional rotation of 0.002 rad/in. This induces a maximum tensile stress of 11,348 psi in the rectangular bar and allows for an additional torsion of the member of 2837 in-lb. For this analysis the bar was assumed to be a uniformly loaded cantilever.

2. Deflection of an Instrument Tip Due to Rotation of the Lower Frame Member

Considering an instrument protruding 60 in. below the frame; the tip deflection, for a design load of 1500 lb,

can be found by applying Equations (18) and (23). The longest unsupported span in the lower frame is 30 in. The loading is considered to be centrally located on the member. The rotation of the member, 4.24×10^{-5} rad/in, yields a tip deflection of 0.0382 in. This added to the deflection due to the instrument drive deformation, Section A-4-a, APP. A, yields a total design tip deflection of 0.17 in. A deflection of this magnitude is not expected to affect the operation of the instruments in any way.

3. Bending of the Lower Frame Members Due to Transverse Loading

The lower frame members, for load limit design purposes, can be considered as beams rigidly fixed at both ends and spanning 100 in. If the load is considered to be applied at the center of the beam, the following equation provides for the maximum moment [7, p. 112].

$$M_{max} = \frac{1}{8} PL \qquad (24)$$

where;

P = transverse load (lb)

L = length of beam, 100 in.

Applying Equation (7) yields the following equation for the maximum load supported by the beam.

$$P = \frac{8\ \sigma_y\ I}{L\ y} \tag{25}$$

where;

y = distance to the outer fiber (in.)

σ_y = yield strength, 40,000 psi for 6061-T6 aluminum

I = moment of inertia of the cross section, Fig. 10, (in^4)

L & P = previously defined

Solving for I

$$I = \frac{1}{4}\pi(R^4 - r^4) + \frac{1}{12}b(H^3 - h^3)$$

where;

R = outside radius of pipe (in.)

r = inside radius of pipe (in.)

b = width of rectangular bar (in.)

$H = 2R + 3.0$

$h = 2R$

$$I = \frac{1}{4}\pi((4.0)^4 - (3.75)^4) + \frac{1}{12}(1.0)((11.0)^3 - (8.0)^3)$$

$$I = 114.0\ in^4$$

Solving for maximum transverse load;

$$P = \frac{8.0(40 \times 10^3)(114.0)}{100(5.5)}$$

$$P = 66{,}300 \text{ lb}$$

With a safety factor of 3, the maximum allowable transverse load is 2.21×10^4 lb. The design load is 1500 lb.

4. Strength of the Bolt Pattern of the Connecting Plates

a. Shearing of the Bolt Pattern

The connection plate, Fig. II-1-1, has eight 9/16 - 16 UNF K-Monel bolts, torqued to 161 in-lb (Sect. A-2-c, APP. A). The bolt preload yields a total compressive load between the plates of 11,491 lb. The stress area of the bolts is 0.203 in^2 each [8]. Failure of the connection consists of slipping of the plates relative to each other and the resulting shearing of the bolts. If the pressure area is assumed to be centered around the bolt centers, the following equation provides the torque required to fail the joint. The coefficient of friction for aluminum to aluminum is 1.1 [15, p. F-16]. The ultimate stress of K-Monel in shear is 98×10^3 psi [8, p. 566].

$$T = R(\mu P + A_s\, \sigma_{ys}) \tag{26}$$

where;

T = torque to fail (in-lb)

R = radius of bolt centers, 5.5 in. (Fig. II-1-1)

μ = coefficient of friction

P = compressive load (lb)

A_s = stress area of bolts (in^2)

σ_{ys} = ultimate strength of the bolts in shear (psi)

Solving;

$$T = 5.5(1.1(11,491.2) + 0.203(98 \times 10^3))$$

$$T = 179,000 \text{ in-lb}$$

With a safety factor of 3, the allowable torque per connection is 59,646 in-lb. This is twice the design torque of 29,250 in-lb (Fig. 8).

b. Bending Moment Required to Open the Bolted Joint

Opening of the connection plates in bending constitutes a mode of failure of the joint. Figure II-1-1 exhibits the bolt diagram of the plate. If the plate is oriented such that point "A" is at the vertical and the joint is assumed to hinge, in failure, about a line connecting the two bolt holes opposite of point "A", the following equation provides the bending moment required to open the joint. The 9/16 in. bolts, torqued to 161 in-lb, provide individual

forces, assumed to be located at the bolt centers, of 1436.4 lb. The bolts act in pairs and the moment arms, from top to bottom, are respectively; 10.16, 7.19, 2.98 in.

$$M_B = 2(1436.4)(10.16 + 7.19 + 2.98)$$

$$M_B = 58,400 \text{ in-lb}$$

If one considers a beam, fixed at both ends, and 100 in. long, the following equation provides for the moment between one end and the centrally applied load [7, p. 112].

$$M = \frac{1}{8} P (4x - L) \tag{27}$$

where;

M = moment at x (in-lb)

x = distance from the end (in.)

P = transverse load (lb)

L = length of the beam (in.)

Solving for P, when the moment at x is the bending moment for failure of the joint. The joint is located 8.25 in. from the end.

$$P = \frac{8(-58,400)}{4(8.25) - 100}$$

$$P = 6972 \text{ lb}$$

If the loading is assumed to be the design insertion load, the joint has a safety factor of 4.6.

c. Strength of the Joint in Tension

Failure of the joint in tension is constituted by the parting of the connection plates. In order for the plates to part, the preload of the eight 9/16 in. bolts must be exceeded. The preload is 1436.4 lb, Section A-2-c, App. A, and the subsequent load required for failure in tension is 11,491 lb.

5. Buckling of the Middle Support Columns

The upper support section can be considered as a column loaded in compression with both ends fixed (Fig. II-2). The following equation provides for the critical buckling load of such a column [14, p. 67].

$$P_{cr} = \frac{4 \pi^2 E I}{L^2} \tag{28}$$

where;

P_{cr} = critical buckling load (lb)

E = modulus of elasticity, 10×10^6 psi

I = moment of inertia of the cross-section (in^4)

L = length of the column, 59.0 in.

Solving;

$$P_{cr} = \frac{4\pi^2(10 \times 10^6)}{(59.0)^2}\left(\frac{1}{4}\pi((4.0)^2 - (3.75)^2)\right)$$

$$P_{cr} = 172{,}600 \text{ lb}$$

This buckling load allows for a vertical load on top of the platform of approximately 5.7×10^5 lb. This is well in excess of any imaginable loading.

6. Determination of the Cut for Piece II-3-4

The piece in question is a part of the lower end section of the main frame (Fig. II-3). It serves as a connection between the middle support and lower end sections. Since this pipe intersects the lower end section at the corner joint of two pipes at an angle of 55.1° from the plane of the lower frame, the cut required on the connecting piece is not a standard curve.

The cut was determined graphically (Fig. II-3-5, fold out) by passing a vertical plane, parallel to the axis of symmetry, through the subject joint. The resulting section was then constructed in plan view and the location of the intersection of piece II-3-4 and the lower frame, relative to a suitable reference frame, was recorded. The reference

frame chosen was an orthogonal frame with two axes in the plane of the section, one of which was parallel to the centerline of piece II-3-4. This process was repeated systematically by passing the section plane at multiples of a short distance, measured perpendicular to the plane of symmetry, until the sections passed out of the realm of the subject joint. The resulting data represents isolated points of intersection between piece II-3-4 and the lower frame. These points can be used to generate a pattern for cutting by determining the circumferential distance between sections and locating the intersections axially on the piece, measuring from a transverse plane. Figure II-3-4 represents the resulting pattern. This pattern is for half the circumference of the pipe, since the pipe is symmetric about a vertical plane. The pattern need only be reversed to mark the opposite side of the pipe.

Since the process excludes the thickness of the pipe, some grinding of the inside of the pipe will be necessary for a close fit. An exact fit is not necessary since the piece is to be welded to the lower frame.

This procedure can be used in any circumstance where an accurate section can be constructed. The accuracy of the method is dependent on the accuracy of the construction of the section and the required measurements. If the thickness of the pipe was included in the constructed section, the shape of the cutting surface for an exact cut could be determined.

C. DESIGN CALCULATIONS FOR THE CORING ASSEMBLY

The coring assembly consists of a coring cylinder (Fig. III-1), the ballast tank and core cylinder support (Fig. III-2), eight cores (Fig. III-3), and the instrument drive to core connector (Fig. III-4-1). Figures (III-5-3,4) show cross sections of the coring assembly. The only loads on the assembly, that are of any consequence, are those experienced during insertion and withdrawal of the core and during blowing of the ballast tank with pressurized air.

1. Shearing of the Core Lifting Bearing Retaining Bolt

The weakest link in the core insertion and withdrawal system is the 5/8-18 UNF bolt attaching the lifting bearing to the core (Fig. III-3-4). The subject bearing is loaded radially during insertion and withdrawal and transmits a shearing load to the bolt. The stress area of the bolt is 0.2555 in^2 [11, P. I-203]. The ultimate stress in shear of the bolt material, K-Monel, is 98×10^3 psi [8, p. 566]. The following equation was used to calculate the ultimate load that the bolt will withstand in shear.

$$P_{max} = \sigma_{us} A_s = 98 \times 10^3 (0.2555)$$

$$P_{max} = 25{,}000 \text{ lb}$$

This load, in addition to the load required to move the inner race of the bearing relative to the core, provides the ultimate load that the bearing connection will withstand.

Since an instrument drive assembly is used to insert and withdraw the core, the maximum load that the bolt will be required to support is the design load of 1500 lb. The subject bolt, due to shear strength alone, has an additional safety factor of over 16.

2. <u>Preload of the Core Lifting Bearing Retaining Bolt</u>

Applying Equations (9) and (10) to the 5/8-18 UNF bolt, the following results are obtained.

Equation (9)

$$Fe = \frac{\sigma_y \ (A_s)^{3/2}}{6}$$

$$Fe = \frac{(111 \times 10^3)(0.2555)^{3/2}}{6}$$

$$Fe = 2390 \text{ lb}$$

Equation (10)

$$T = C \ D \ Fe$$

$$T = 0.20(5/8)(2390)$$

$$T = 299 \text{ in-lb}$$

The maximum torque allowable is 299 in-lb. Since the bolt preload is required only to clamp the inner race of the bearing, to prevent rotation, this maximum is not required.

A preload of 500 lb, requiring a torque of 62.5 in-lb is recommended as a satisfactory specification.

3. Stresses in the Ballast Tank During Ballast Blowing

The ballast tank can be considered as a circular cylinder 82 in. long with a mean radius of 11.75 in. and a shell thickness of 0.5 in. (Fig. III-2). One end is closed by a circular plate, 0.5 in. thick, and the other end is open. The ballast water will be blown with 15 psig air.

a. Hoop Stresses in the Ballast Tank

The hoop stress in the cylinder due to the internal pressure, in areas not affected by the end conditions, is given by the following equation [17, p. 398]. The pressure used for the calculation is 45 psig, a safety factor of 3 over the actual value.

$$\sigma_t = \frac{P\ a}{h} \tag{29}$$

where;

P = 45 psig

a = mean radius, 11.75 in.

h = thickness, 0.5 in.

solving;

$$\sigma_t = \frac{45(11.75)}{0.5}$$

$$\sigma_t = 1057 \text{ psi}$$

The increase in radius of the cylinder due to the internal pressure is given by the following equation [17, p. 398].

$$\delta = \frac{a\ \sigma_t}{E} \tag{30}$$

where;

a = mean radius, 11.75 in.

E = elastic modulus, 10×10^6 psi, 6061-T6 aluminum

σ_t = hoop stress (psi)

solving;

$$\delta = \frac{11.75(1057)}{10 \times 10^6}$$

$$\delta = 1.25 \times 10^{-3} \text{ in.}$$

This is the major outward deflection of the cylinder during operation.

b. Stress Due to the Restraining Ring

The following equation is used to determine the moment in the cylinder, beneath the ring (Fig. III-2-5), [17, p. 405-406].

$$M_o = \frac{P(1 - \frac{1}{2}\nu)}{2\beta^2} x_2(2\alpha) \tag{31}$$

where;

$$\beta = [\frac{3(1 - \nu^2)}{a^2 h^2}]^{1/4}$$

$$x_2(2\alpha) = \frac{\text{Sinh } 2\alpha - \text{Sin } 2\alpha}{\text{Sinh } 2\alpha + \text{Sin } 2\alpha}$$

P = pressure, 45 psig

ν = Poissons ratio, 0.33 [8, p. 566]

a = mean radius, 11.75 in.

h = thickness, 0.5 in.

L = length between restraints, 70 in.

$\alpha = \frac{\beta L}{2}$

solving;

$$\beta = [\frac{3(1 - (0.33)^2)}{(11.75)^2(0.5)^2}]^{1/4}$$

$$\beta = 0.5275$$

$$\alpha = \frac{0.5275(70)}{2}$$

$$\alpha = 18.46$$

$$x_2(2\alpha) = \frac{\text{Sinh}(36.92) - \text{Sin}(36.92)}{\text{Sinh}(36.92) + \text{Sin}(36.92)}$$

$$x_2(2\alpha) = 1.00000$$

$$M_o = \frac{45(1 - \frac{1}{2}(0.33))}{2(0.5275)^2}(1.000)$$

M_o = 67.5 in-lb/in of circumference

The maximum stress is the tensile stress beneath the ring due to the moment and the closed end of the cylinder. Equation (7) in addition to the effects of the end load, provides the following.

$$\sigma = \frac{M_o 6}{h^2} + \frac{P a}{2h}$$

where;

M_o = moment (in-lb)

P = pressure, 45 psig

a = mean radius, 11.75 in.

h = thickness, 0.5 in.

solving;

$$\sigma = \frac{67.5(6)}{(0.5)^2} + \frac{45(11.75)}{2(0.5)}$$

$\sigma = 2149$ psi

The material to be used is 6061-T6 aluminum with a yield stress of 40,000 psi.

c. Stress Due to the End Closure

The dislocation moment and shear due to the end being closed by a flat plate are determined by the following method [7, p. 307].

$$M_o = \frac{A + B/C}{2\beta + F - G/C} \tag{32}$$

$$V_o = M_o(2\beta + F) - A \tag{33}$$

where;

$$A = \frac{P\ a^3\ \beta^2}{4(1 + \nu)}$$

$$B = \frac{2\ P\ a^2\ \beta^3\ D}{(1 - \frac{1}{2}\nu)}$$

$$C = Eh + 2aD\beta^3(1 - \nu)$$

$$F = \frac{2\ a\ \beta^2}{1 + \nu}$$

$$G = \beta\ E\ h$$

$$D = \frac{E\ h^3}{12(1 - \nu^2)}$$

$$\beta = [\frac{3(1 - \nu^2)}{a^2\ h^2}]^{1/4}$$

E = modulus of elasticity, 10×10^6 psi

ν = Poissons ratio, 0.33

P = pressure, 45 psig

a = mean radius, 11.75 in.

h = thickness, 0.5 in.

solving;

$$\beta = [\frac{3(1 - (0.33)^2)}{(11.75)^2(0.5)^2}]^{1/4}$$

$$\beta = 0.5275$$

$$D = \frac{10 \times 10^6\ (0.5)^3}{12(1 - (0.33)^2)}$$

$$D = 1.1690 \times 10^5$$

$$G = 0.5275(10 \times 10^6)(0.5)$$

$$G = 2.638 \times 10^6$$

$$F = \frac{2(11.75)(0.5275)^2}{1 + 0.33}$$

$$F = 4.9166$$

$$C = 10 \times 10^6(0.5) + 2(11.75)(1.169 \times 10^5)(0.5275)^3(0.67)$$

$$C = 5.270 \times 10^6$$

$$B = \frac{2(45)(11.75)^2(0.5275)^3(1.169 \times 10^5)}{1 - \frac{1}{2}(0.33)}$$

$$B = 2.5534 \times 10^8$$

$$A = \frac{45(11.75)^3(0.5275)^2}{4(1 + 0.33)}$$

$$A = 3818.2$$

$$M_o = \frac{3818.2 + \dfrac{2.5534 \times 10^8}{5.270 \times 10^6}}{2(0.5275) + 4.9166 - \dfrac{2.638 \times 10^6}{5.270 \times 10^6}}$$

$$M_o = 706.75 \text{ in-lb/in of circumference}$$

$$V_o = 706.75(2(0.5275) + 4.9166) - 3818.2$$

$$V_o = 402.23 \text{ lb/in of circumference}$$

The maximum bending, hoop, and shear stress in the cylinder occur at its end and are provided by the following equations [7, p. 302].

Hoop stress

$$\sigma_h = \frac{2\ \beta\ a}{h}\ (-V_o + M_o\beta) + \frac{P\ a}{h} \tag{34}$$

Bending stress plus longitudinal stress

$$\sigma_b = -\frac{6\ M_o}{h^2} + \frac{P\ a}{2h} \tag{35}$$

Shear stress

$$\tau = \frac{V_o}{h} \tag{36}$$

where;

All variables are as defined before. Solving;

$$\sigma_h = \frac{2(0.5275)(11.75)}{0.5}(-402.23 + 706.75(0.5275))$$

$$+ \frac{45(11.75)}{0.5}$$

σ_h = 328.12 psi, tension

$$\sigma_b = -\frac{6(706.75)}{(0.5)^2} + \frac{45(11.75)}{2(0.5)}$$

σ_b = -16,430 psi, compression

$$\tau = \frac{402.23}{0.5}$$

$$\tau = 804 \text{ psi}$$

The stresses are below the limiting stresses of 6061-T6 aluminum.

The stresses in the circular end plate are equal to or less than those in the cylinder and can be found from equations presented by R. Roark [7, p. 216,219]. The material to be used is 6061-T6 aluminum with a yield strength in tension of 40,000 psi and an ultimate shear strength of 30,000 psi.

4. Torque Required to Rotate the Coring Cylinder While Submerged

When the platform is submerged, the coring cylinder with applied flotation exerts a buoyant force of 565 lb on the top plate of the main frame (Fig. III-1-3) through the coring cylinder bearings (Fig. III-1-6). The coefficient of friction for Delrin AF is 0.10, resulting in a tangential force of 56.5 lb during rotation. If the force is assumed to act through the center of the bearing surface, the resulting torque required to rotate the cylinder is 791 in-lb.

D. DESIGN CALCULATIONS FOR THE MOTOR BOXES

The Motor Boxes are used to provide power to the instrument drive assembly and the coring cylinder. Since the boxes are constructed from commercially available components, the

main design problem is fitting the pieces. The calculations show the acceptability of the chosen gears. Each box is designed to be oil filled and pressure compensated. Windage and friction losses are neglected. The loss in the instrument drive box is small and the coring cylinder box is over powered enough to compensate for its losses. The main criteria for design are expense and ease of assembly and maintenance.

1. Design of the Instrument Drive Motor Box

The instrument drive requires 3/4 HP to operate under design load (Sect. A-7-d, APP. A). A permanent magnet field dc motor was selected on the basis of its desirable weight and size. The motor (Fig. IV-1-1) provides 3/4 HP at 1140 RPM. The drive train consists of a bull and pinion arrangement in order to reduce the output speed to 190 RPM. The allowable tooth load for the gears shown in Fig. IV-1-1 can be calculated from the following equation [8, p. 366].

$$F_s = \frac{\sigma_y \; b \; y}{K_f \; P_d} \tag{37}$$

where;

F = tooth load (lb)

σ_y = yield strength of the material
steel - 30×10^3 psi
cast iron - 12×10^3 psi

b = tooth width (in.)

y = Lewis modulus [4, Table AT-24]
20° pressure angle, mid tooth
16 teeth - 0.503
96 teeth - 0.752

P_d = diametral pitch (in^{-1})

16 teeth/1.333 in. = 12 in^{-1}

96 teeth/8.000 in. = 12 in^{-1}

K_f = 1.7 [8, p. 366]

The pinion gear has 16 teeth and is constructed of steel. Applying equation (37) the following results are obtained.

$$F_s = \frac{30 \times 10^3 (1)(0.503)}{(1.7)12}$$

$$F_s = 740 \text{ lb}$$

Applying equation (37) to the bull gear, 96 teeth and cast iron construction, results in the following.

$$F_s = \frac{12 \times 10^3 (1)(0.752)}{(1.7)12}$$

$$F_s = 442 \text{ lb}$$

The following equation, derived from the typical horsepower equation, provides the tooth load required to transmit a given horsepower.

$$F_M = \frac{HP\ (3.96 \times 10^5)}{\pi\ N\ D_p} \tag{38}$$

where;

F_M = maximum tooth force during mesh, tangent to the pitch diameter circle (lb)

HP = horsepower to be transmitted

N = RPM

D_p = pitch diameter (in.)

note;

This equation can be applied to either of the meshed gears with equal results.

Applying equation (38) to the pinion gear, the following results are obtained.

$$F_M = \frac{0.75\ (3.96 \times 10^5)}{\pi(1140)(1.333)}$$

$$F_M = 62.20 \text{ lb}$$

This represents a safety factor of approximately 7 as compared to the allowable load of the bull gear.

The bearings selected for the drive train have a safety factor of approximately 10 and are expected to operate for the life of the platform.

2. Design of the Coring Cylinder Motor Box

The coring cylinder requires 791 in-lb to rotate at 0.241 RPM (Sect. C-4, APP. A). The power requirement, with a safety factor of three, is 0.009 HP. The output shaft of the motor box rotates at 1.929 RPM. The torque required, at the output shaft, to transmit 0.009 HP is 300 in-lb. A

permanent magnet field dc motor rated 1/12 HP at 2500 RPM drives a four stage gear train of uniform reduction (1:6) per stage. Since all gears are uniform, the critical mesh is at the output reduction. Due to the chosen motor, the maximum loading would occur during adverse operating conditions, such as when the coring cylinder is nearly locked, when 1/12 (0.0833) HP would be transmitted through the gear train. Applying equation (38) to this, the tooth load without the torque limiter would be 726 lb. Applying equation (37), the allowable tooth load is 265 lb. This represents an allowable output torque of 995 in-lb. A torque limiter setting of 900 in-lb provides an additional safety factor of 3.0, relative to the required torque to rotate the coring cylinder, and a safety factor of 1.1, relative to the allowable tooth load. An output torque of 900 in-lb represents a tooth load of 240 lb.

Should a stronger output stage be desired, the gears can be replaced with 14.5° pressure angle, 12 pitch, 0.75 in. face, gears with 15 and 90 teeth for the pinion and bull respectively, Browning No. NSS1215 and NCG1290. These are readily available and will provide the same gear ratio. The allowable load for this gear set is 458 lb, providing an allowable output torque of 1716 in-lb. This provides for a safety factor of 1.7 relative to the allowable tooth load, when the torque limiter is set at 1000 in-lb.

The bearings selected have a safety factor of approximately 3 and are expected to operate for the life of the platform.

3. Justification for the Design Procedure

Prudent gear selection techniques require the calculation and comparison of three basic load limiting parameters for a given gear. The parameters are tooth strength, F_s (Eqn. 37), dynamic tooth load, F_d, and limiting load for wear, F_w. When a gear is designed such that $F_s \geq F_d$ and $F_w \geq F_d$ it will provide continuous indefinite service [8, p. 362-379].

The dynamic tooth load is evaluated from the following equation [8, p. 369].

$$F_d = K\ F_M \tag{39}$$

where;

$$K = \frac{600 + V_M}{600}$$

V_M = pitch line velocity (ft/min)

F_M = from equation (38)

This equation increases the load due to the power being transmitted by a factor dependent on the pitch line velocity and the accuracy of the tooth profiles. As the pitch line velocity approaches zero, F_d approaches F_M.

The limiting load for wear is given by an algebraic equation presented by V. Faires [8, p. 378]. It will not be given due to its length and complexity. F_w is dependent on pressure angle, gear physical dimensions and properties, and

ratio of the gear set. The result is the limiting load for a gear to operate indefinitely under varying service.

Since the motor boxes will not be required to operate indefinitely, under normal design qualifications of indefinite, the limiting load for wear was not considered in the design. In order to accumulate 1 hour of total running time on the gears, the platform would have to be lowered to the sea floor at least 10 times. This represents a miniscule amount of wear when compared to the life of the platform.

The dynamic load criterion was also neglected when comparisons with the results given represented no change in the actual design. The factor, K (eqn. 39), for the instrument drive motor box is 1.663. This, when applied to F_m, has the effect of reducing the safety factor from 7 to 4. The factor, K, for the coring cylinder motor box is 1.006. This would increase the load required to transmit the power by 0.6%. Under conditions of very short service, the tooth strength can be less than the dynamic tooth load [8, p. 379].

APPENDIX B

Engineering Drawings

The following drawings are, for the most part, accurate scale drawings of individual components of the individual systems of the platform. Some liberties have been taken in dimensioning, scaling, and general drawing practices.

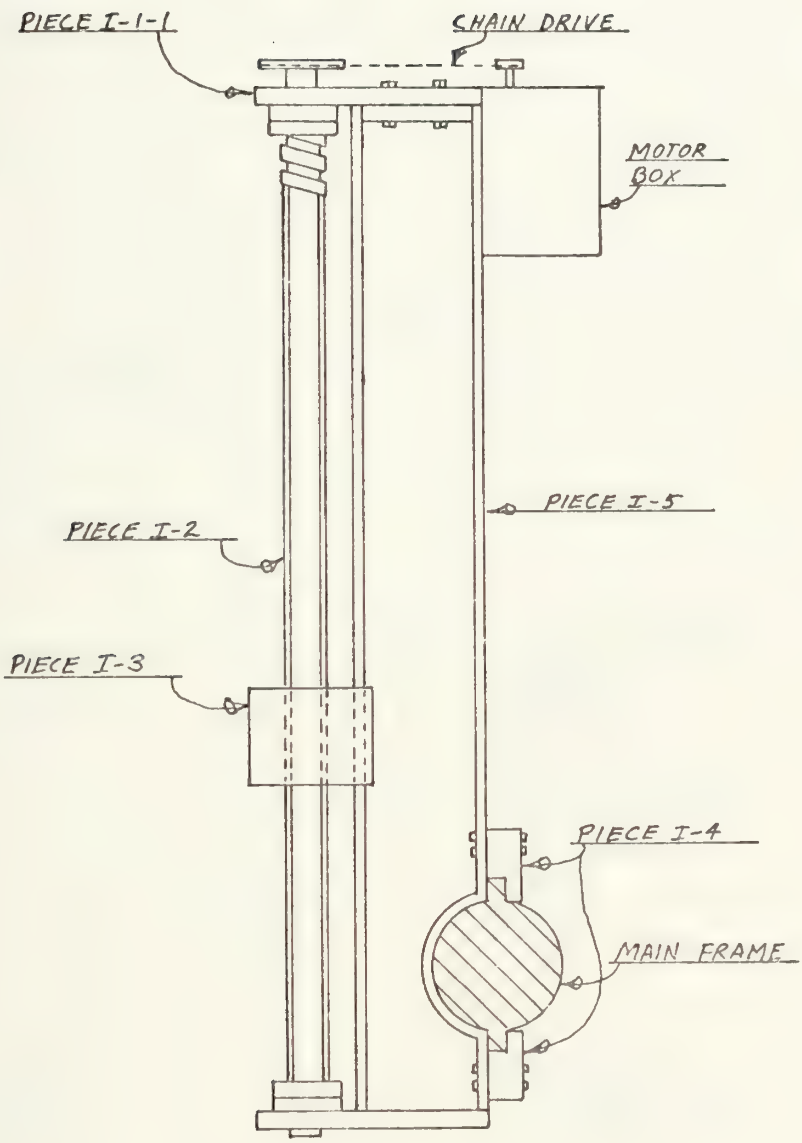


FIG. I. INSTRUMENT DRIVE ASSEMBLY

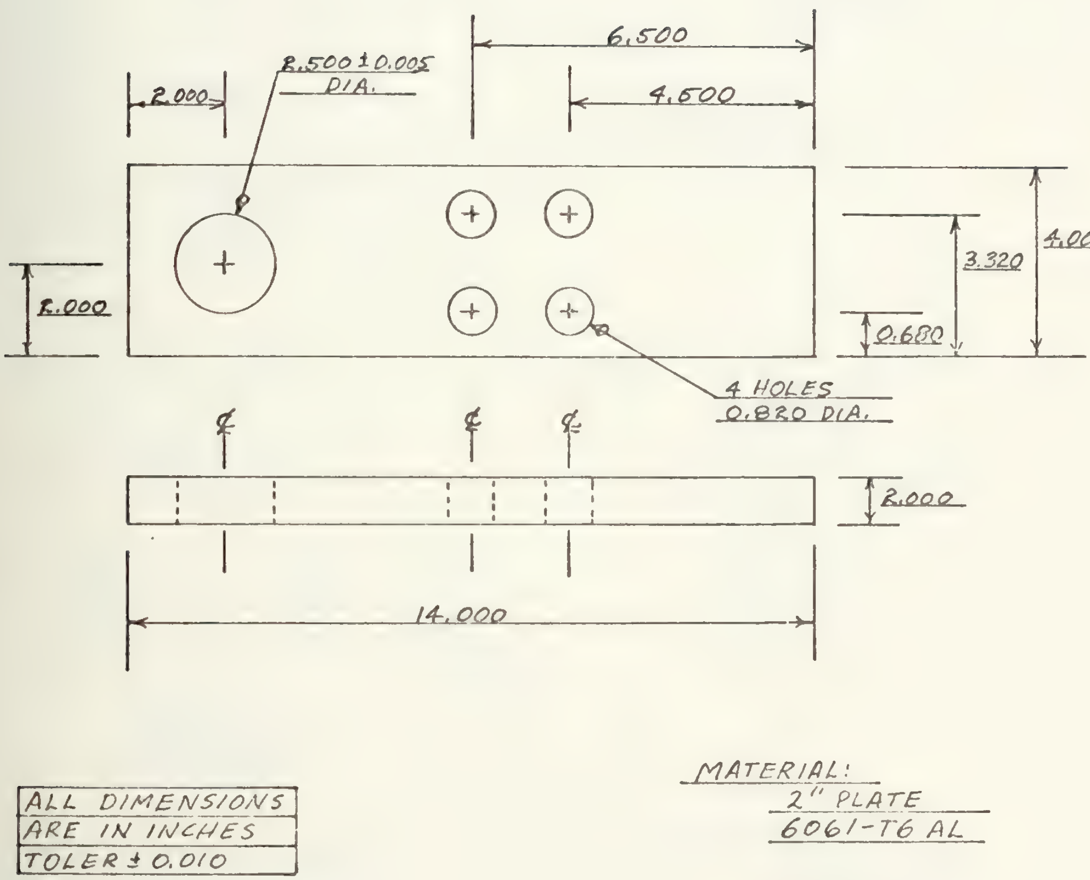


FIG. I-1. PIECE I-1-1, UPPER BEARING PLATE

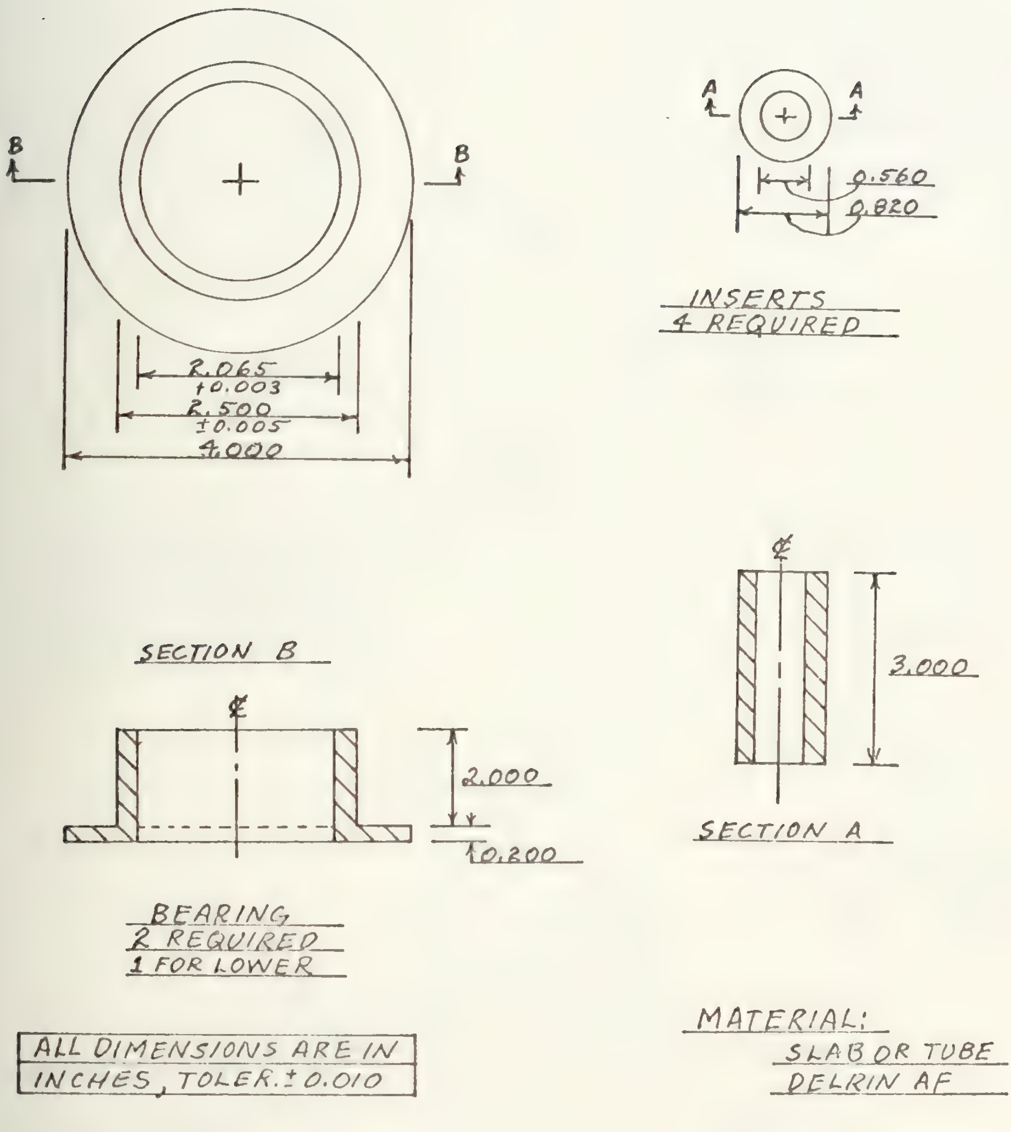


FIG. I-1-1. PIECE I-1-2, BEARINGS AND INSERTS

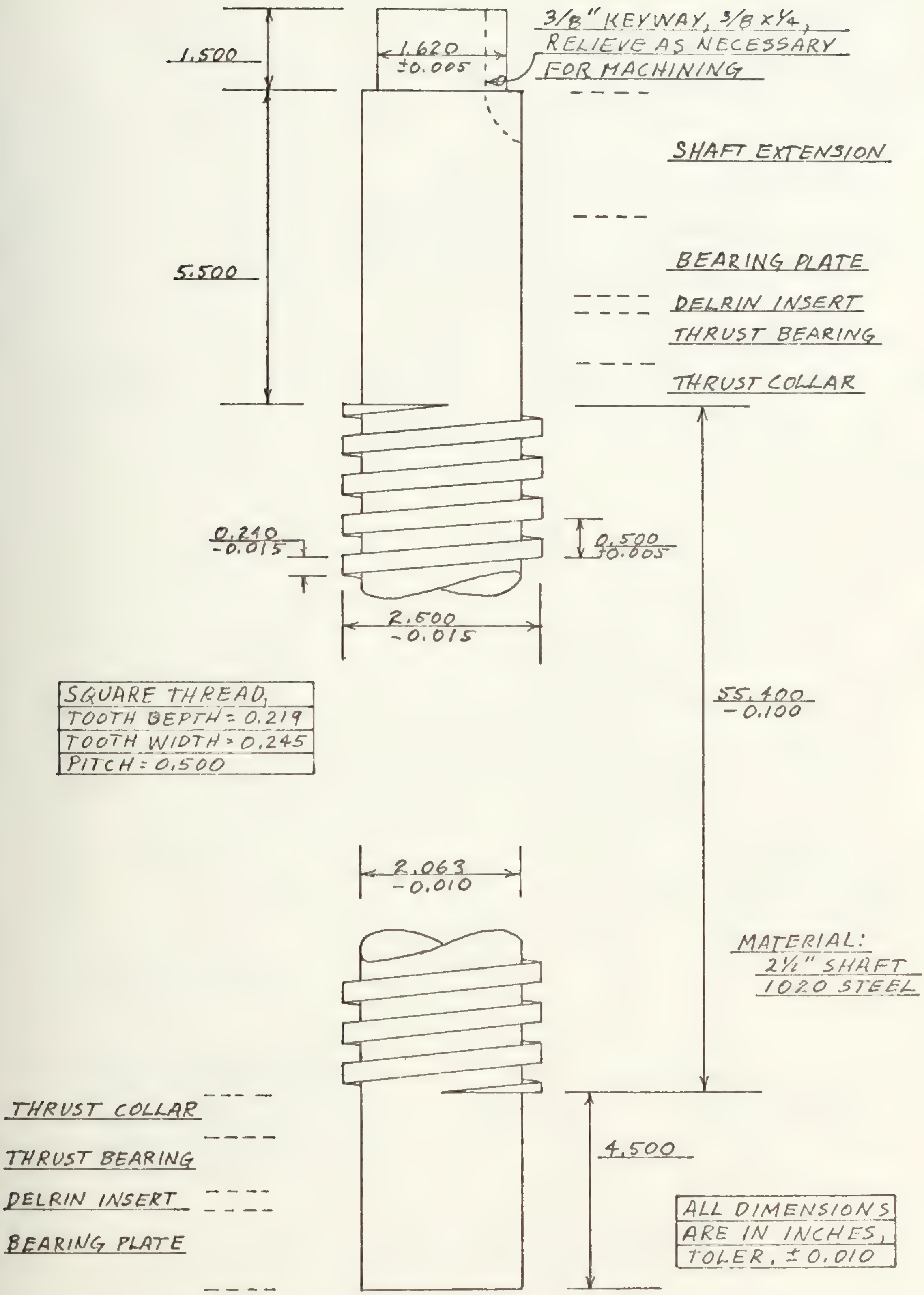


FIG. I-2. PIECE I-2, SCREW SHAFT

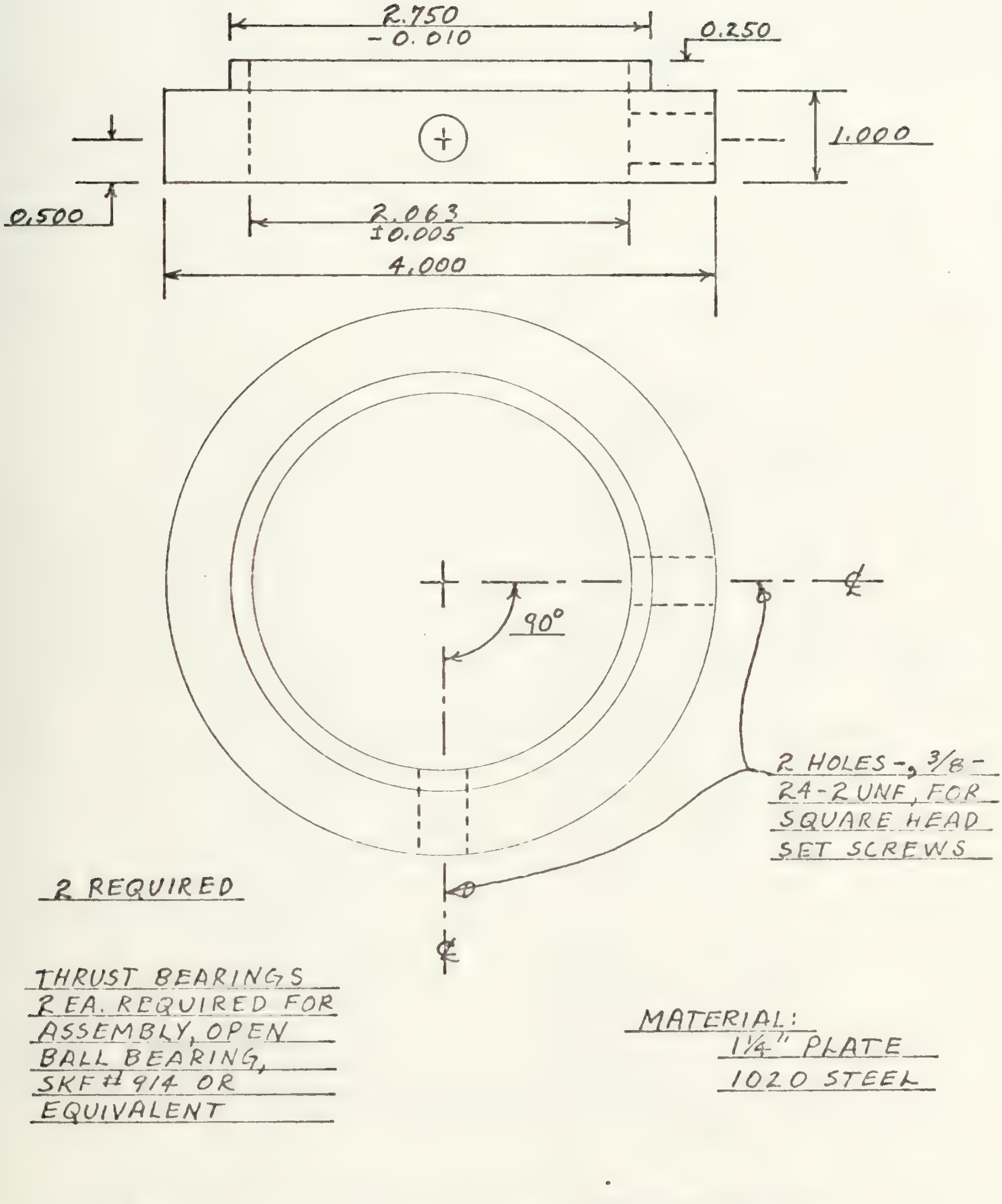


FIG. I-2-1. PIECE I-2-1, THRUST COLLAR

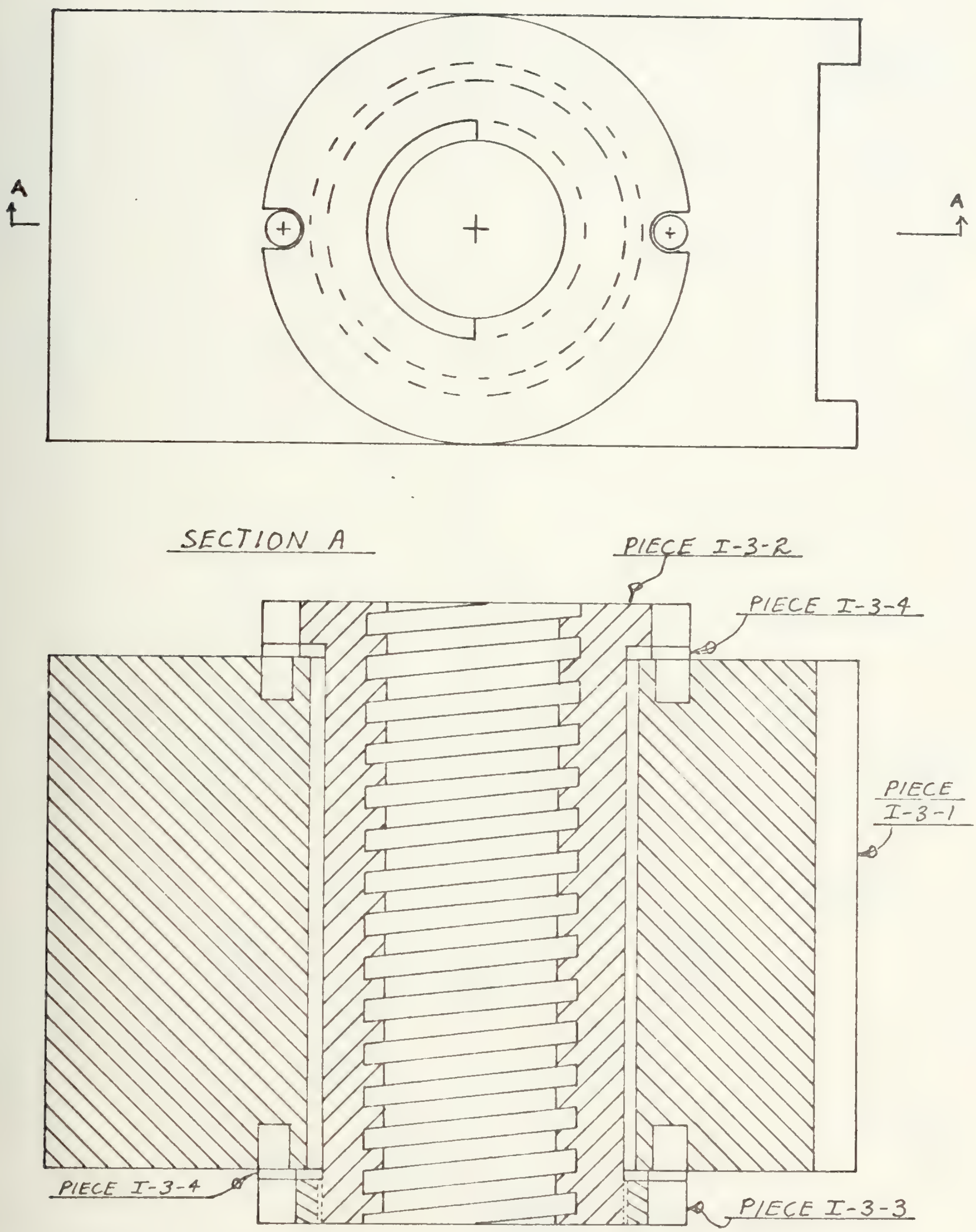


FIG. I-3. PIECE I-3, INSTRUMENT MOUNT BLOCK

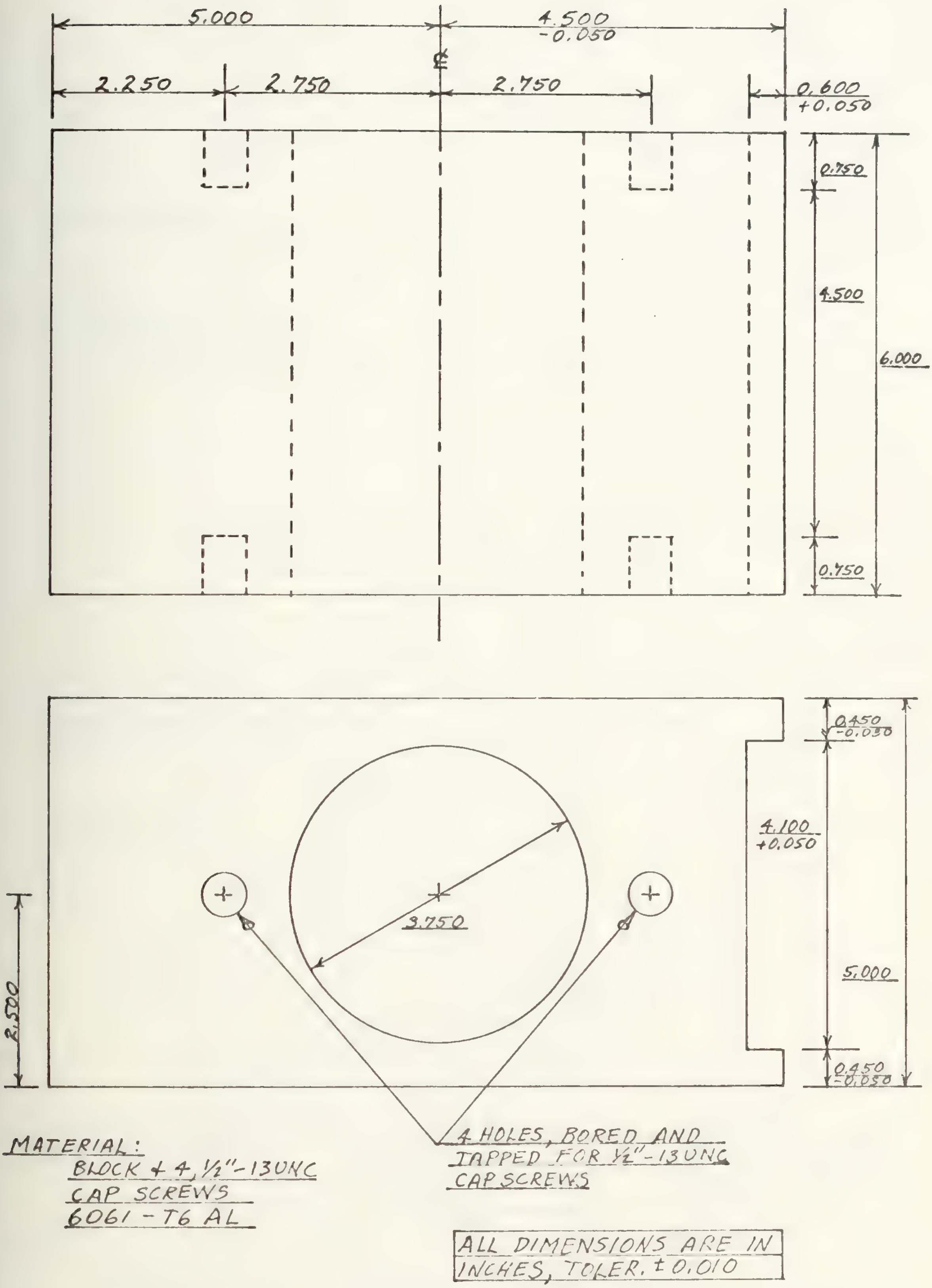


FIGURE I-3-1. PIECE I-3-1, INSTRUMENT BLOCK

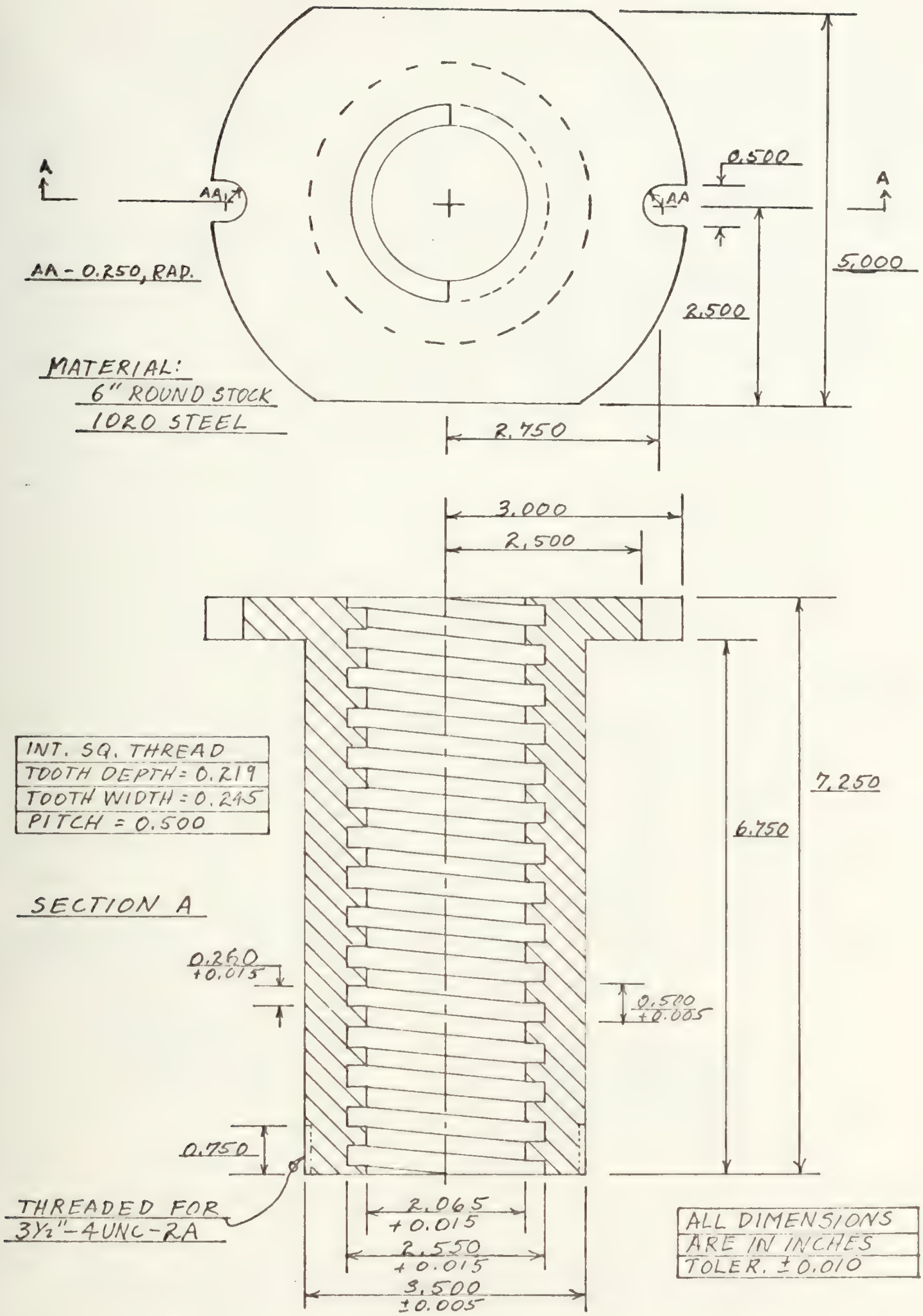


FIGURE I-3-2. PIECE I-3-2, DRIVE CORE

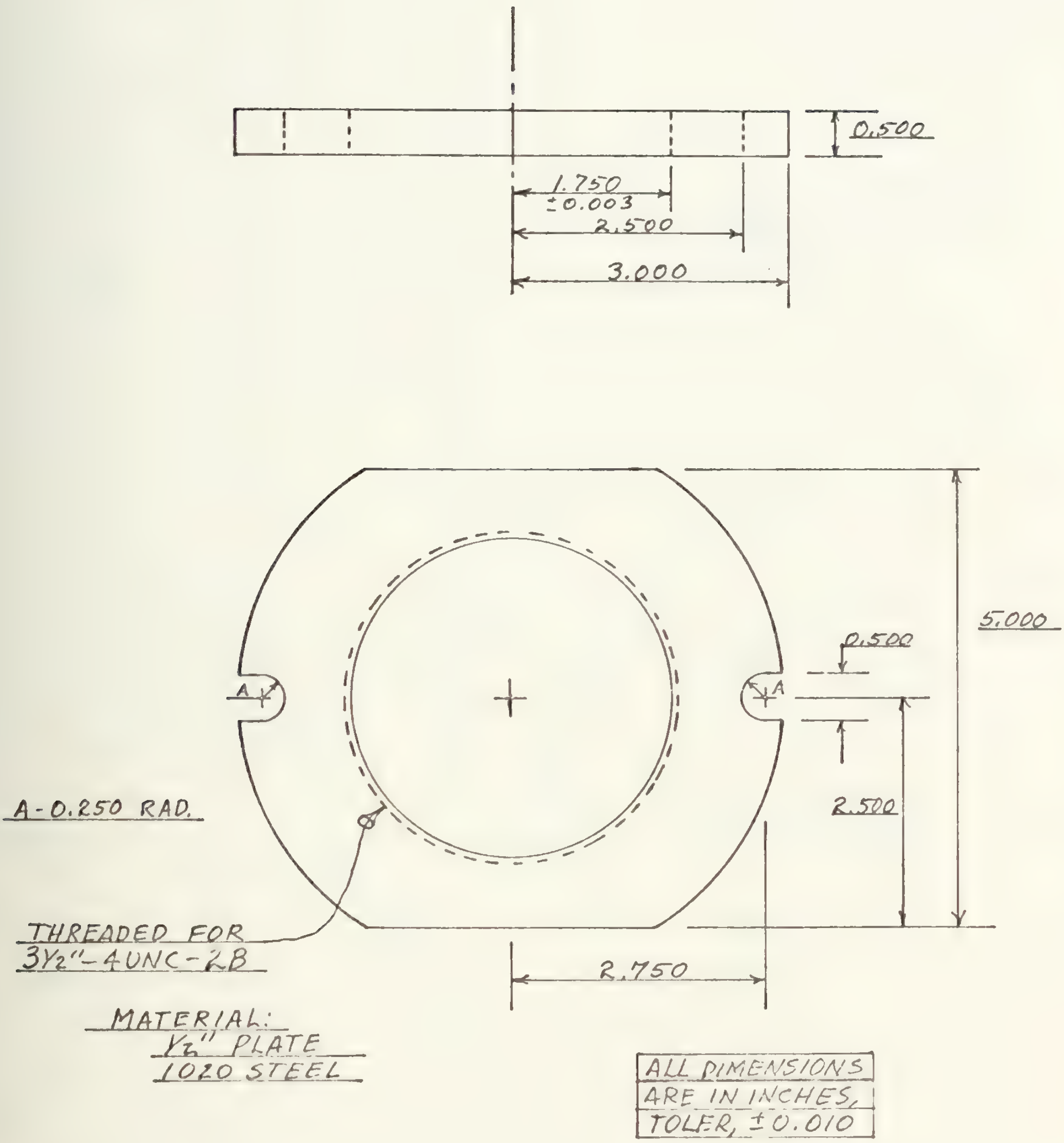


FIGURE I-3-3. PIECE I-3-3, DRIVE CORE NUT

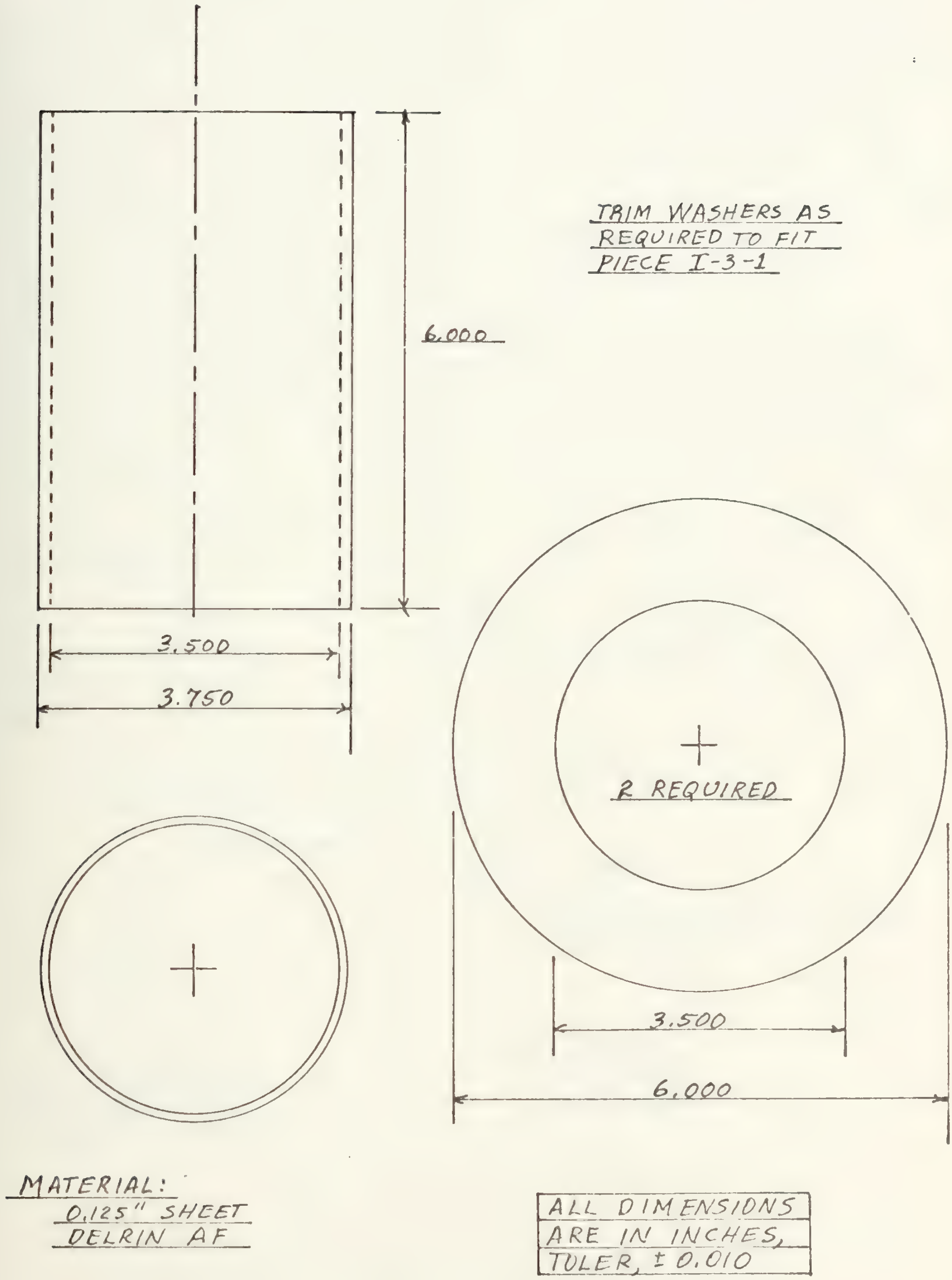


FIGURE I-3-4. PIECE I-3-4, INSULATORS

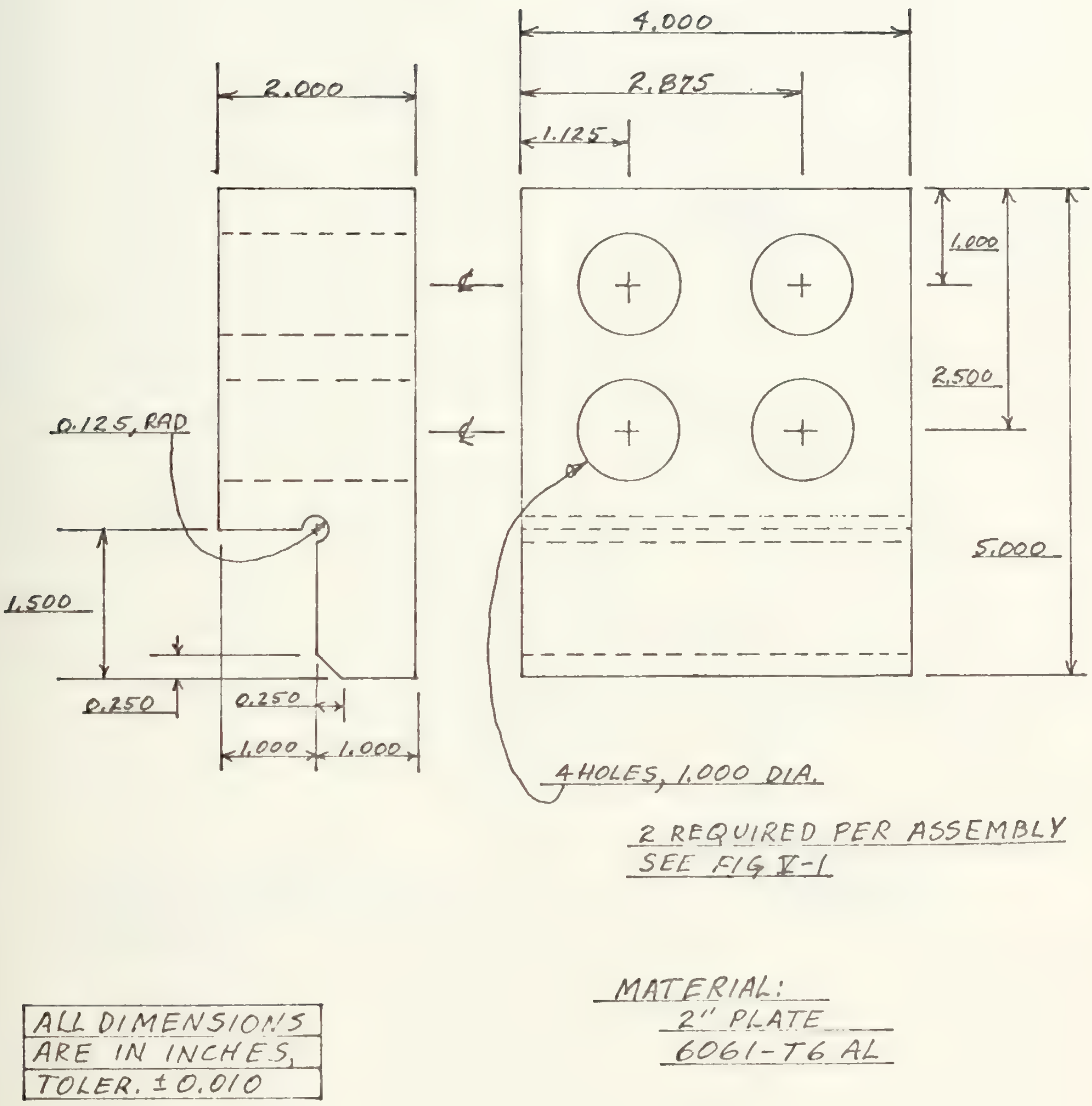


FIGURE I-4. PIECE I-4, MOUNTING BRACKET

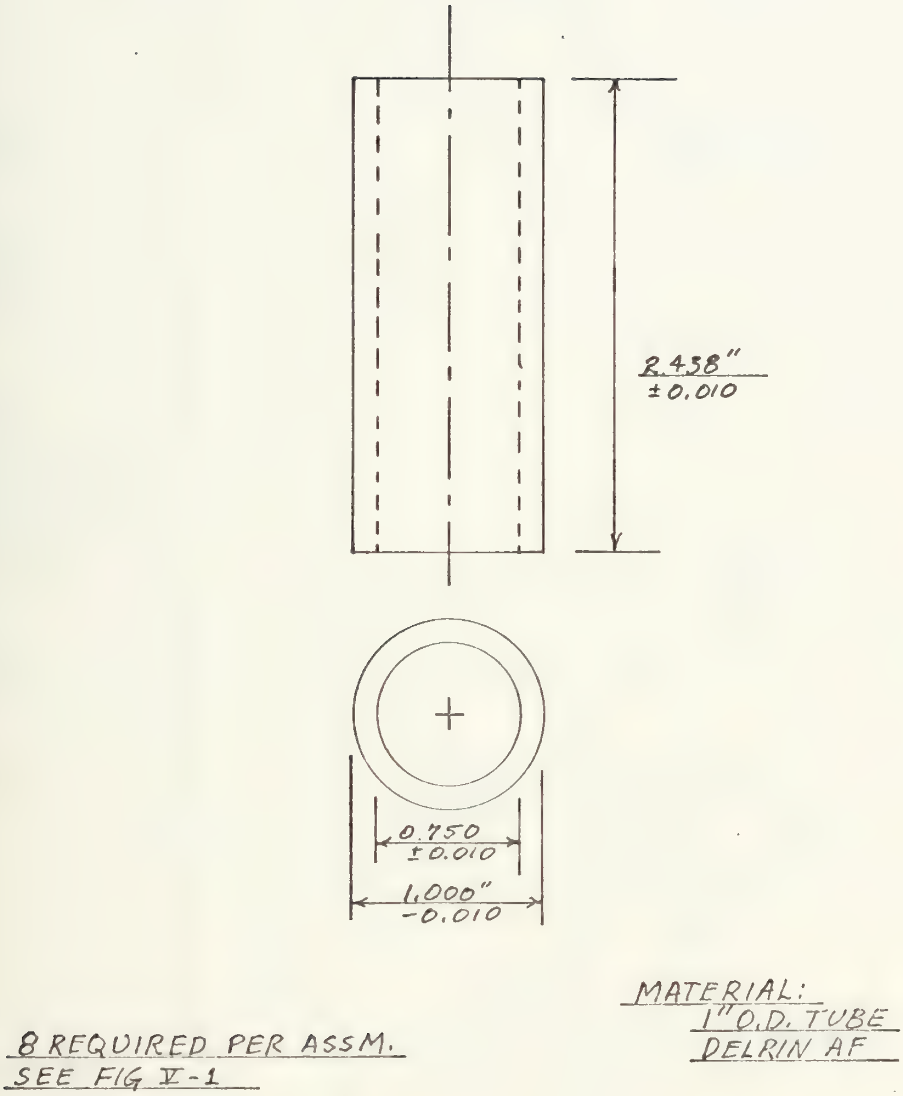


FIGURE I-4-1. PIECE I-4-1, MOUNTING BRACKET
BOLT INSULATORS

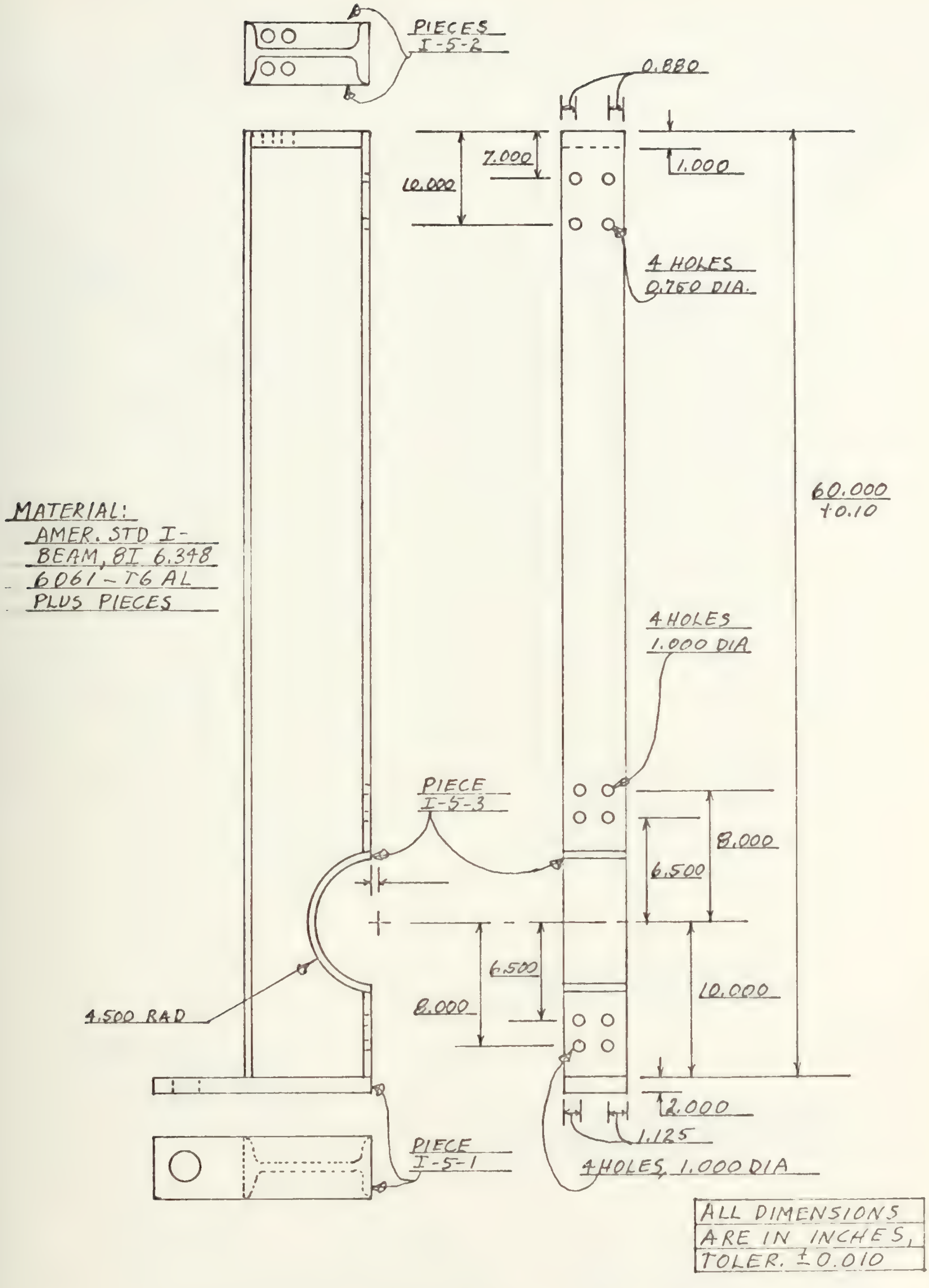


FIGURE I-5. PIECE I-5, INSTRUMENT DRIVE FRAME

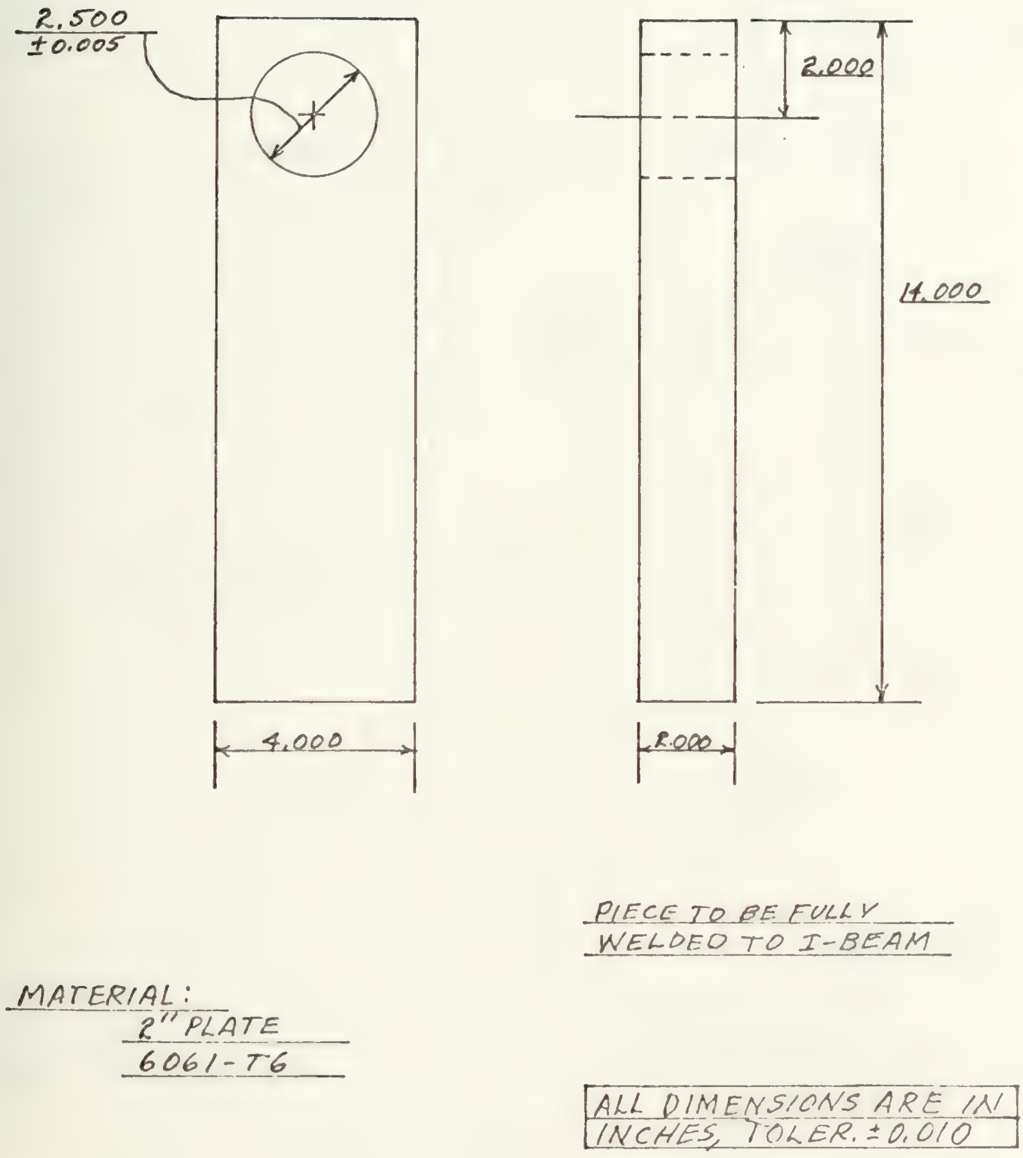


FIGURE I-5-1. PIECE I-5-1, LOWER BEARING PLATE

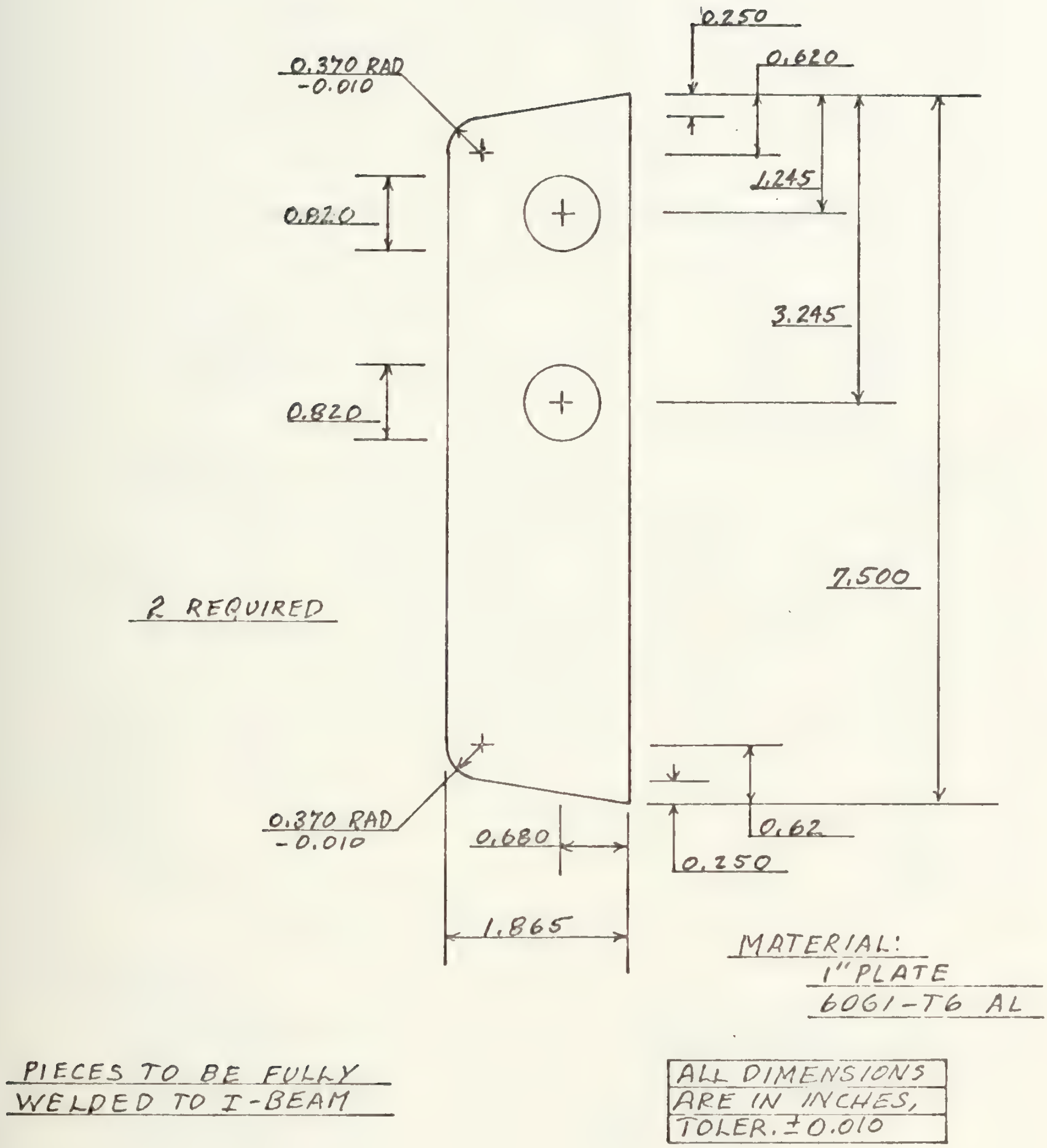


FIGURE I-5-2. PIECE I-5-2, BOLT PLATES

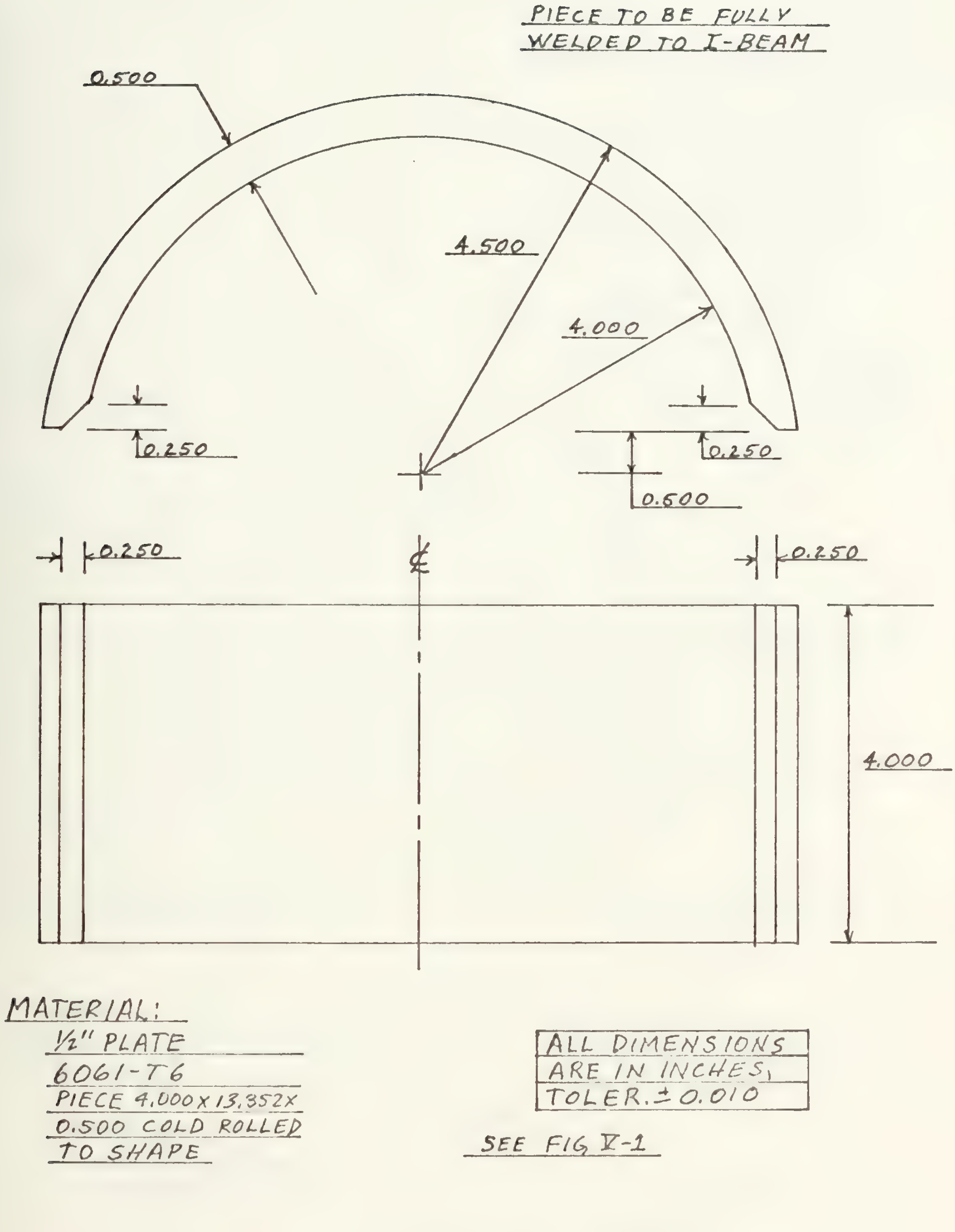


FIGURE I-5-3. PIECE I-5-3, PLATE

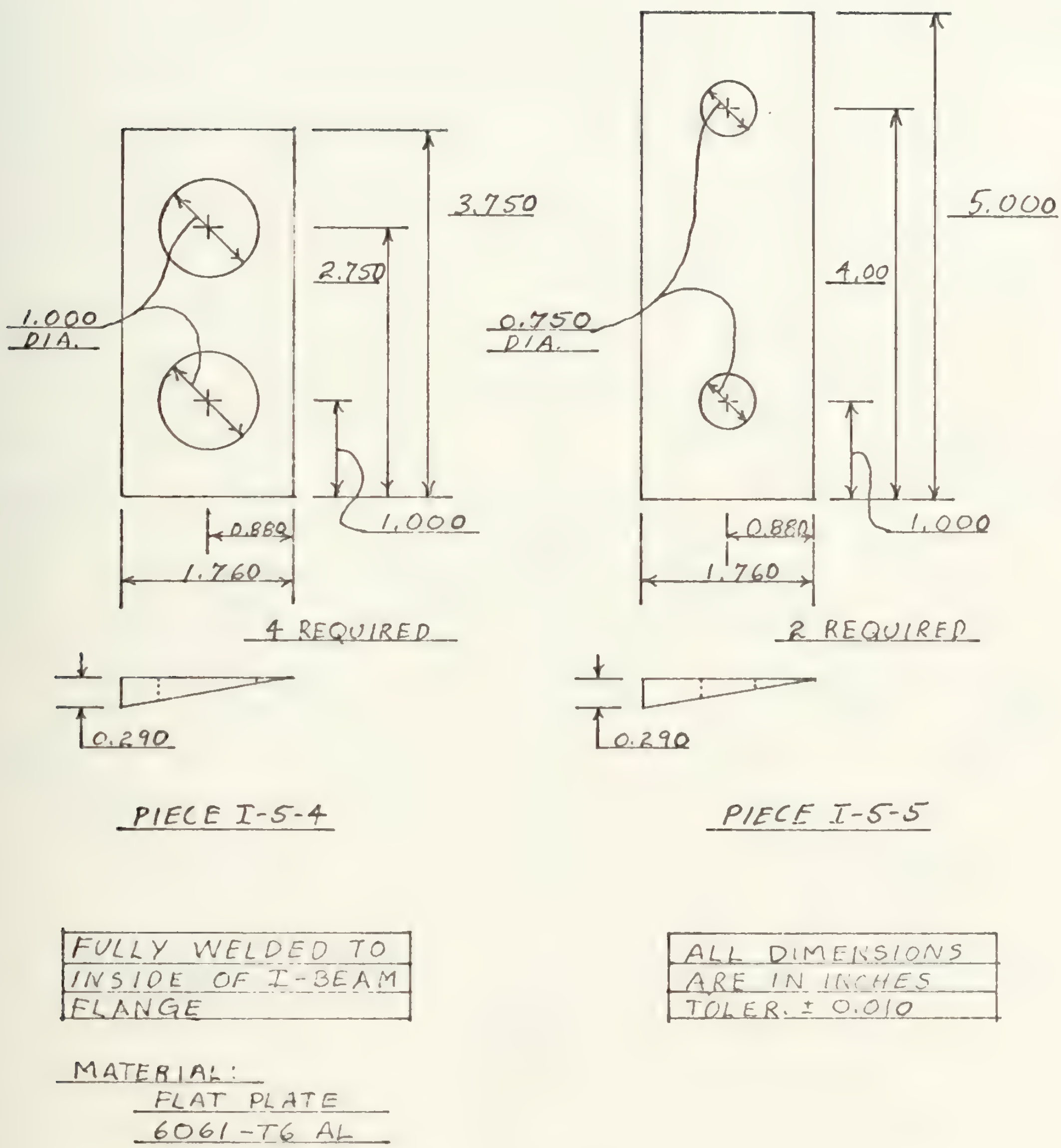


FIGURE I-5-4. PIECES I-5-4,5, BOLT SPACERS

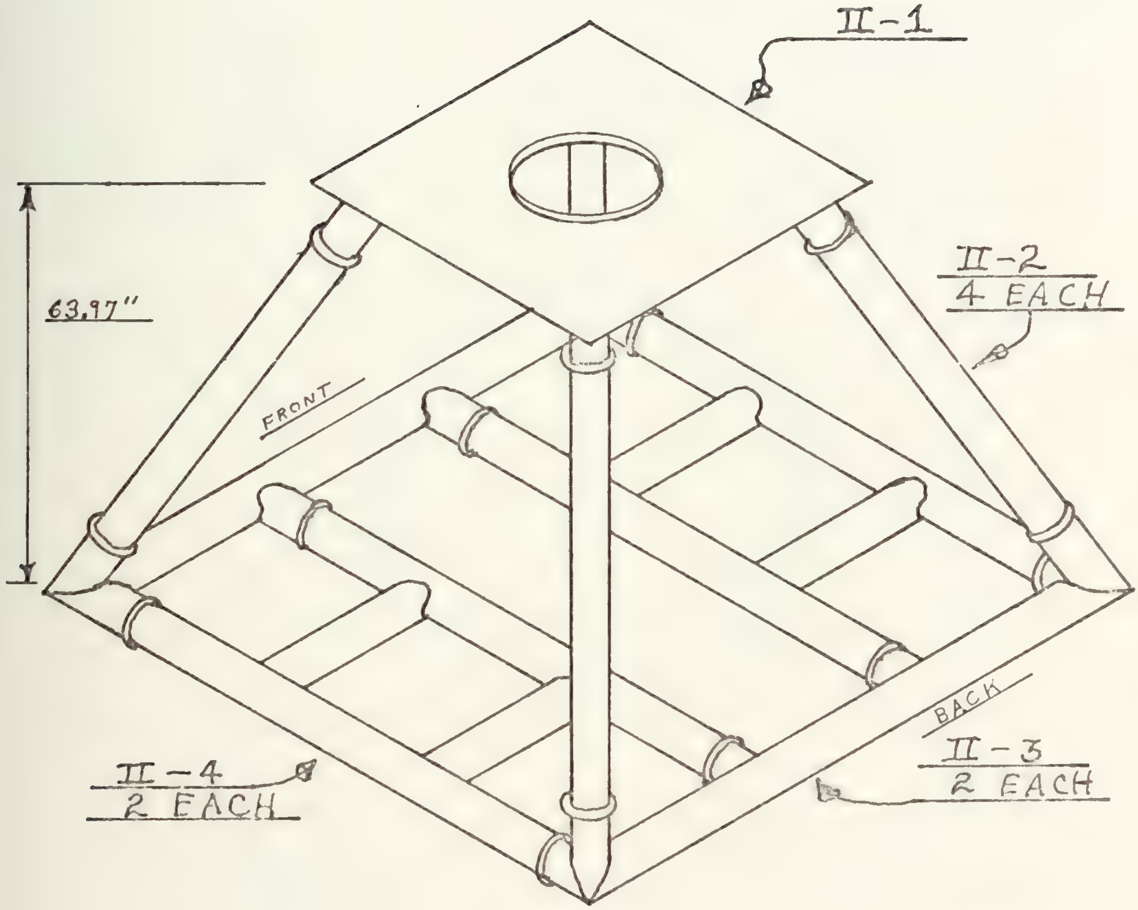


FIGURE II. MAIN FRAME

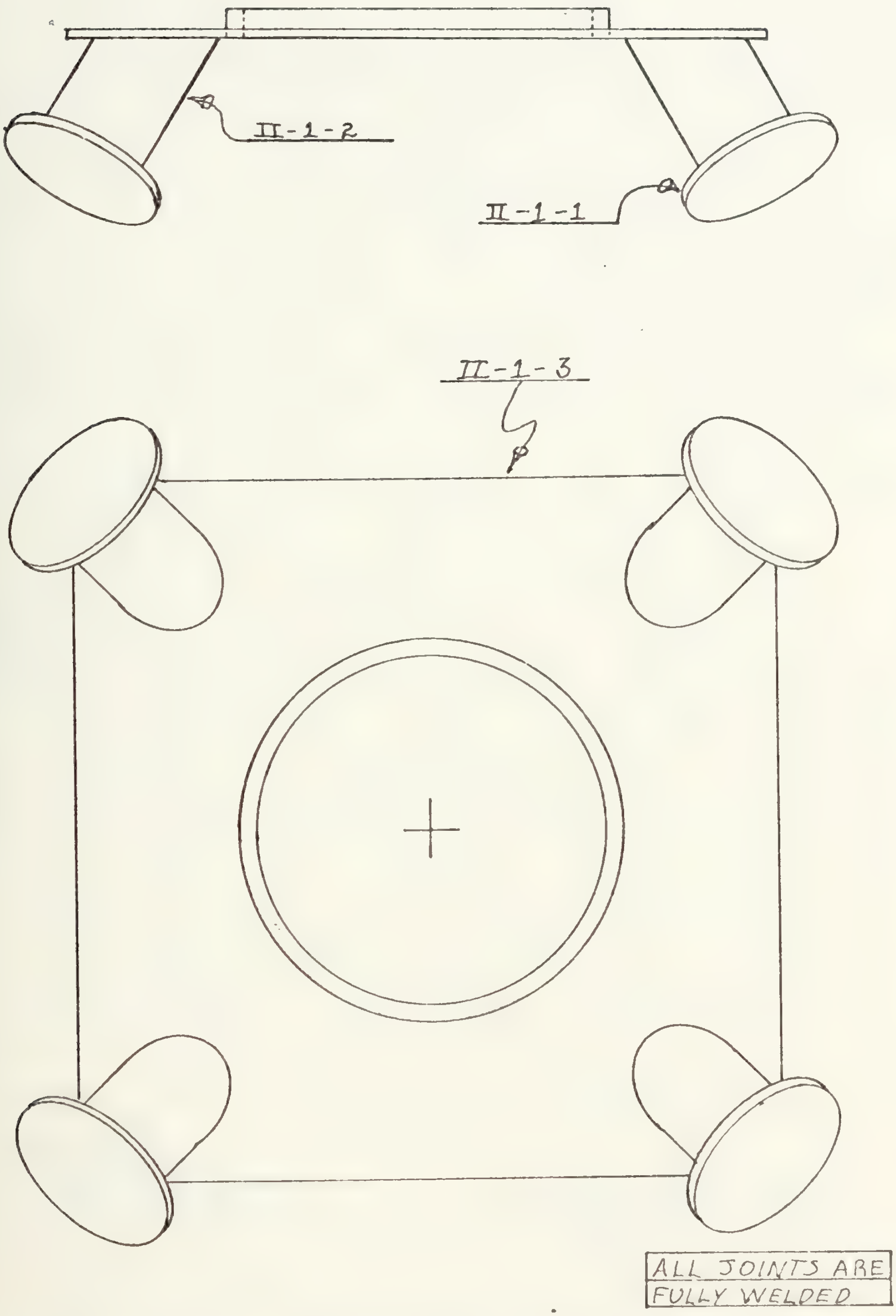


FIGURE II-1. PIECE II-1, MAIN FRAME TOP PIECE

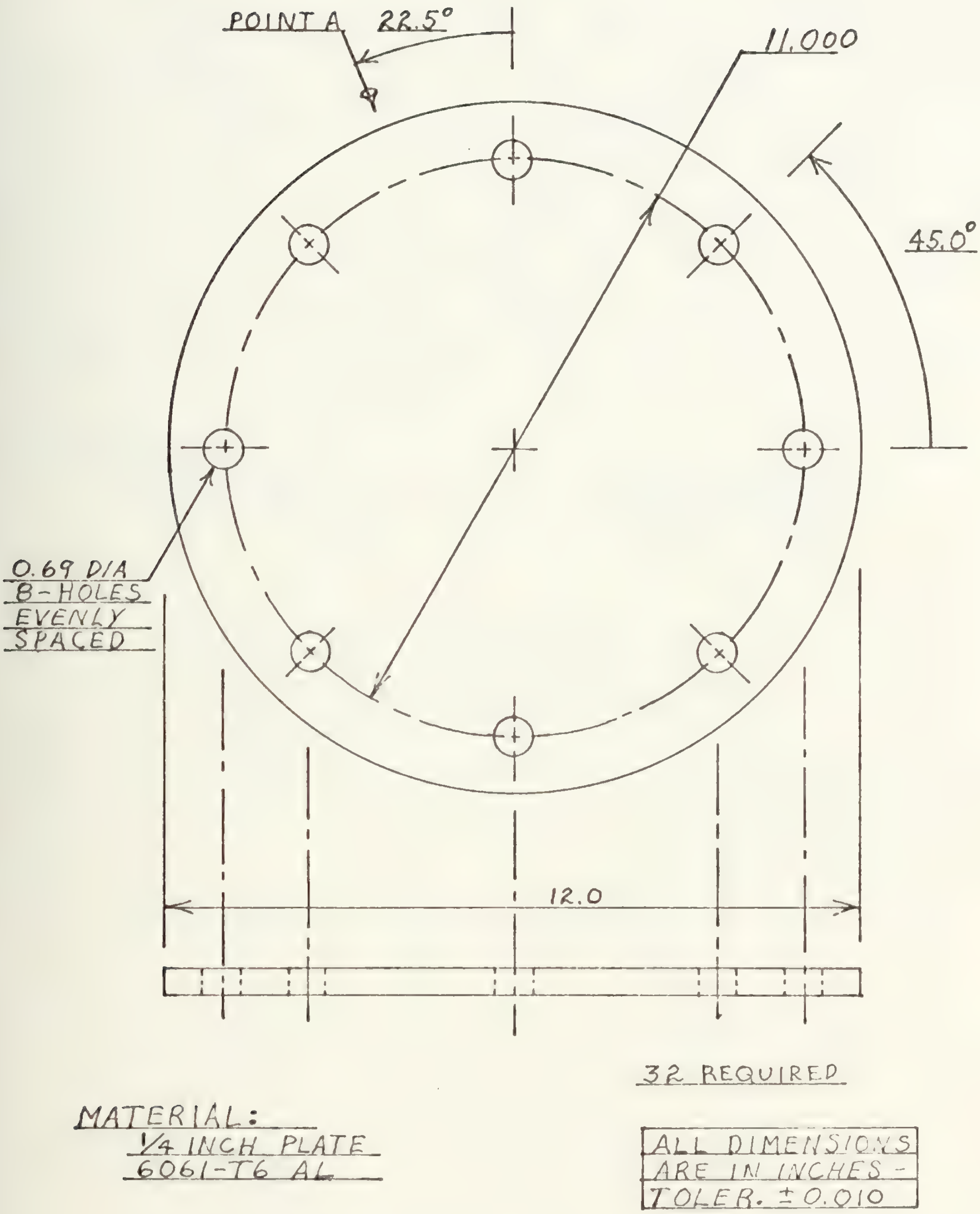


FIGURE II-1-1. PIECE II-1-1, CONNECTION PLATE

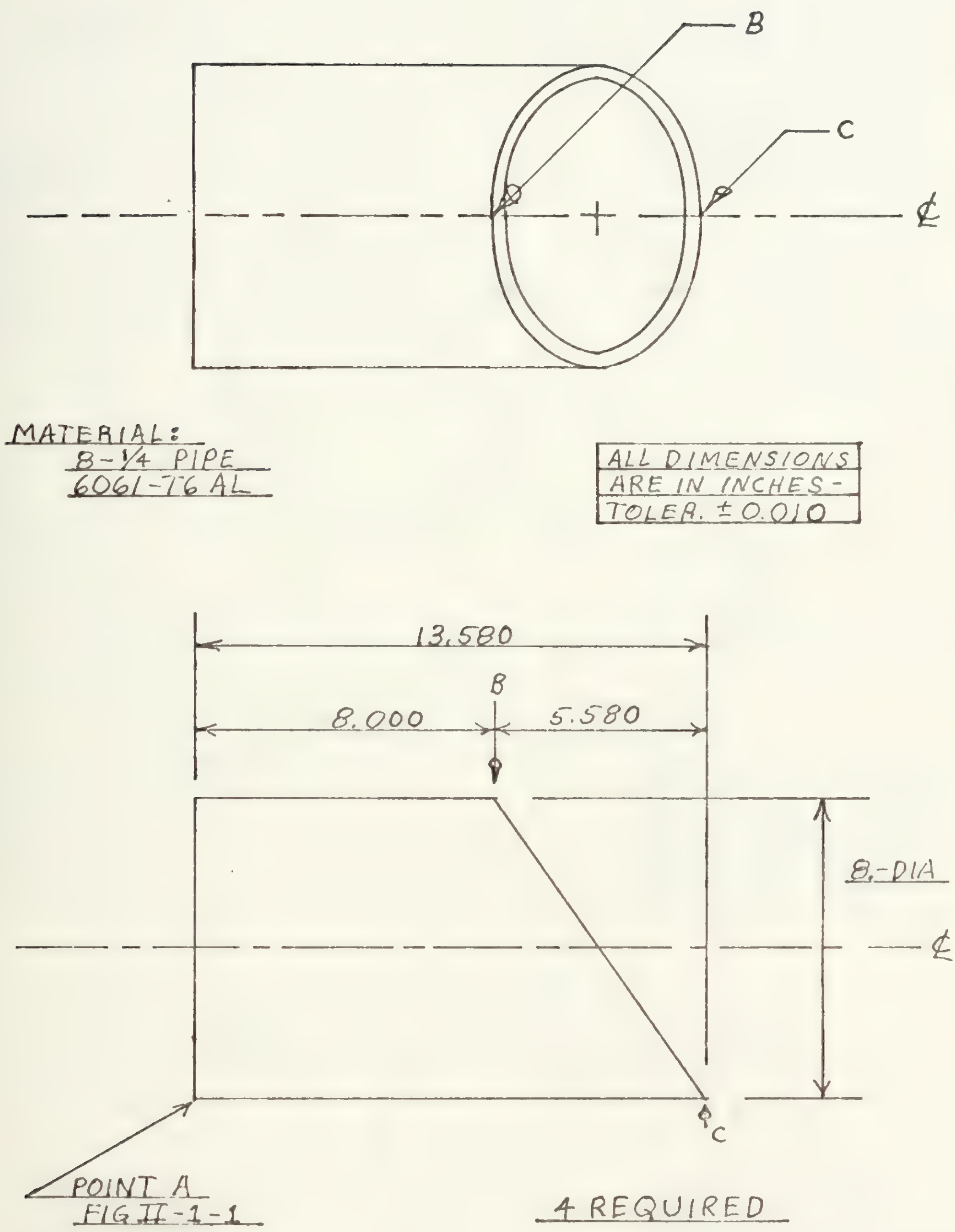


FIGURE II-1-2. PIECE II-1-2, UPPER SUPPORT SECTION

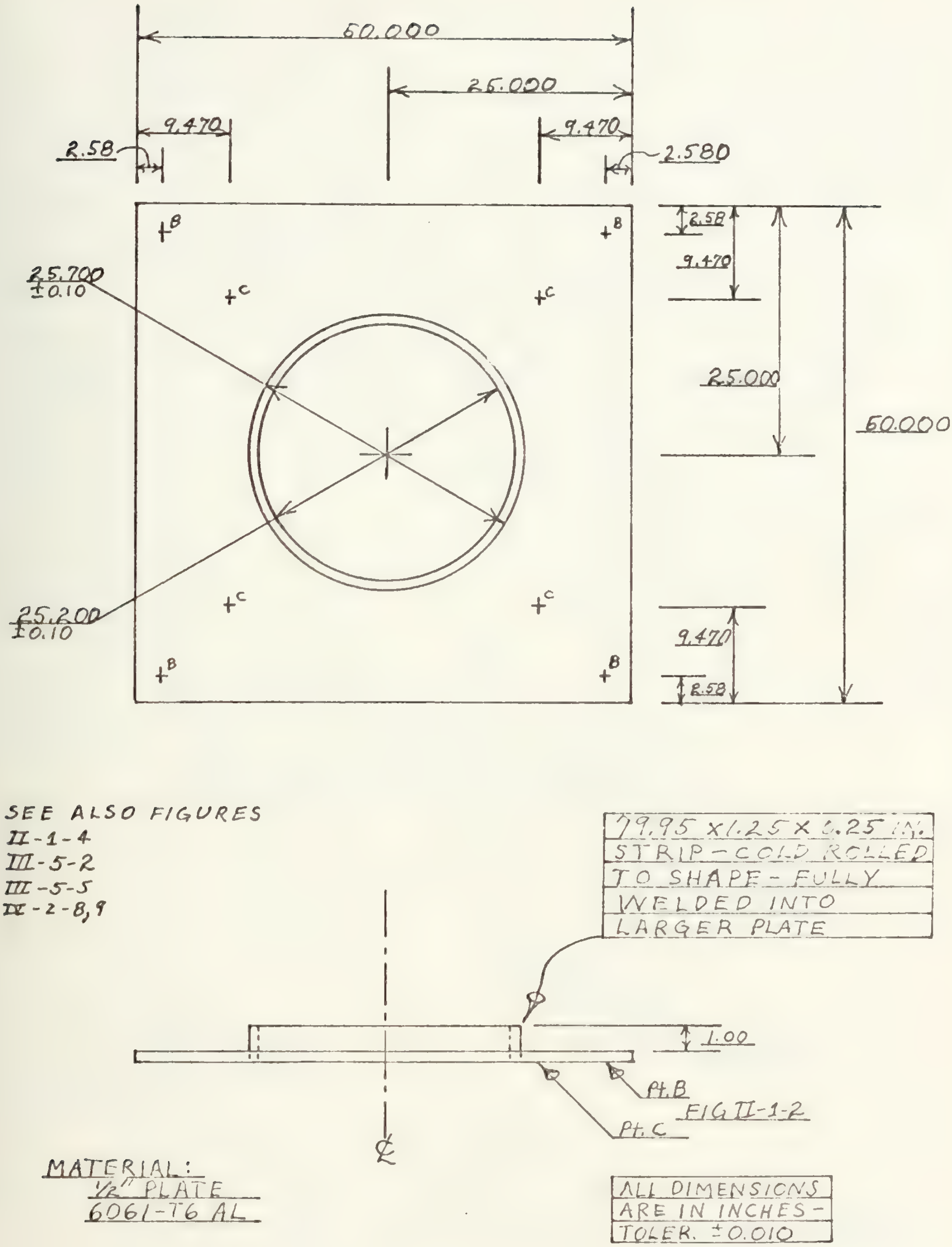
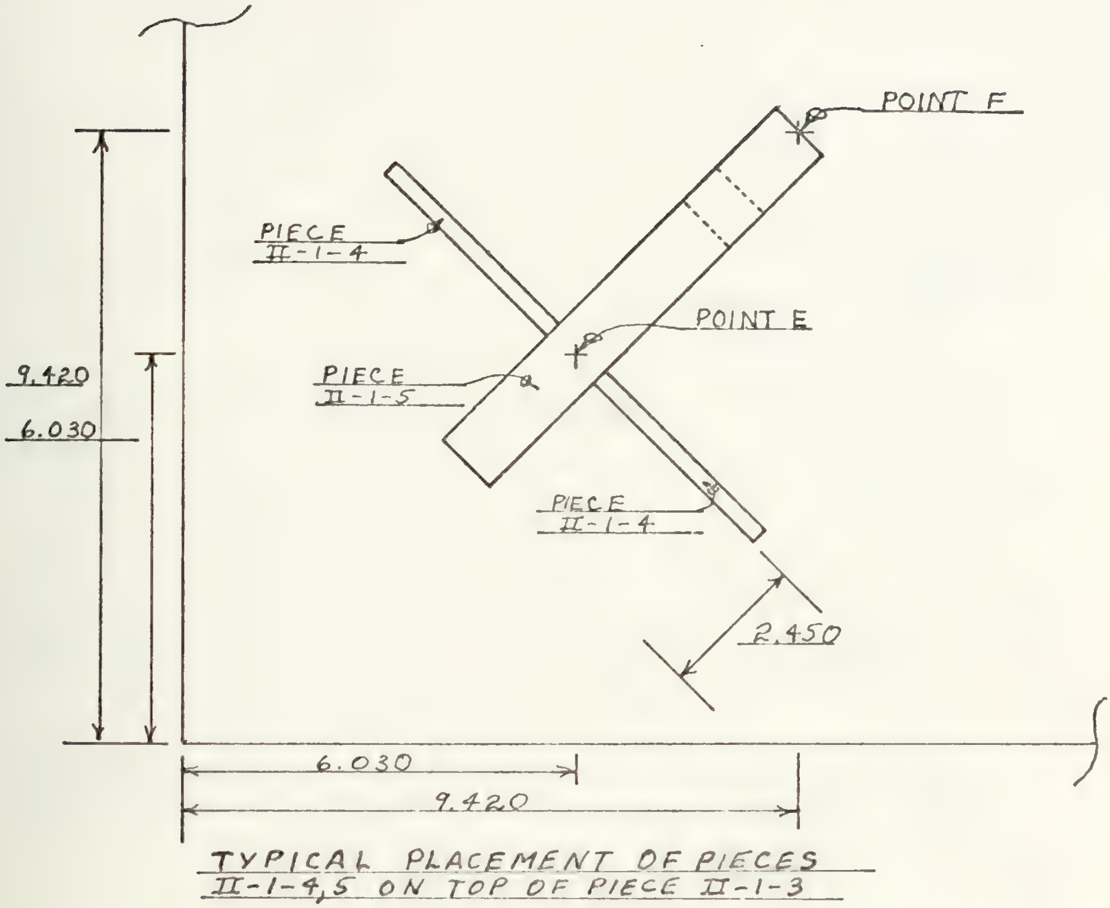


FIGURE II-1-3. PIECE II-1-3, TOP PLATE



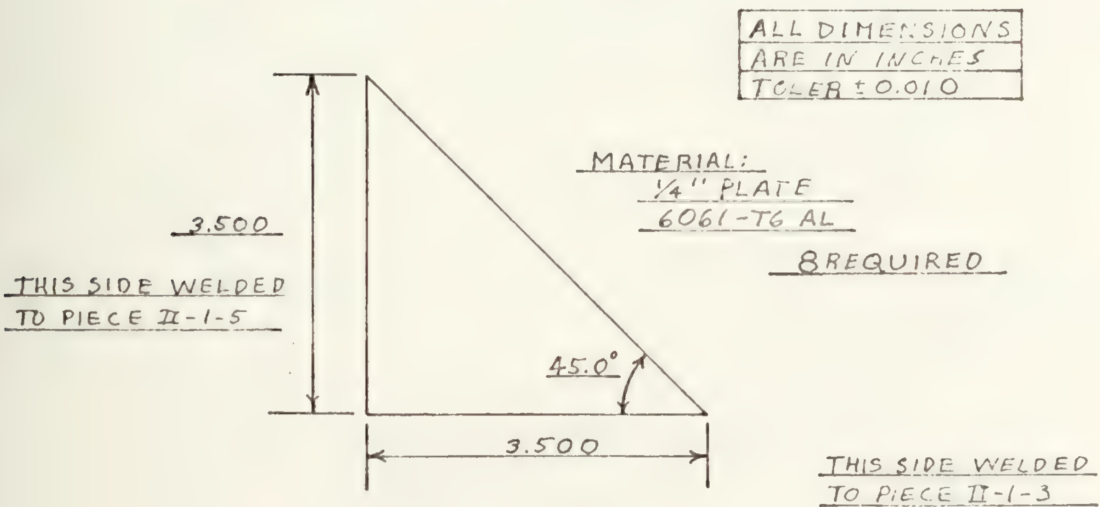


FIGURE II-1-4. PIECE II-1-4, LIFTING PAD BRACE

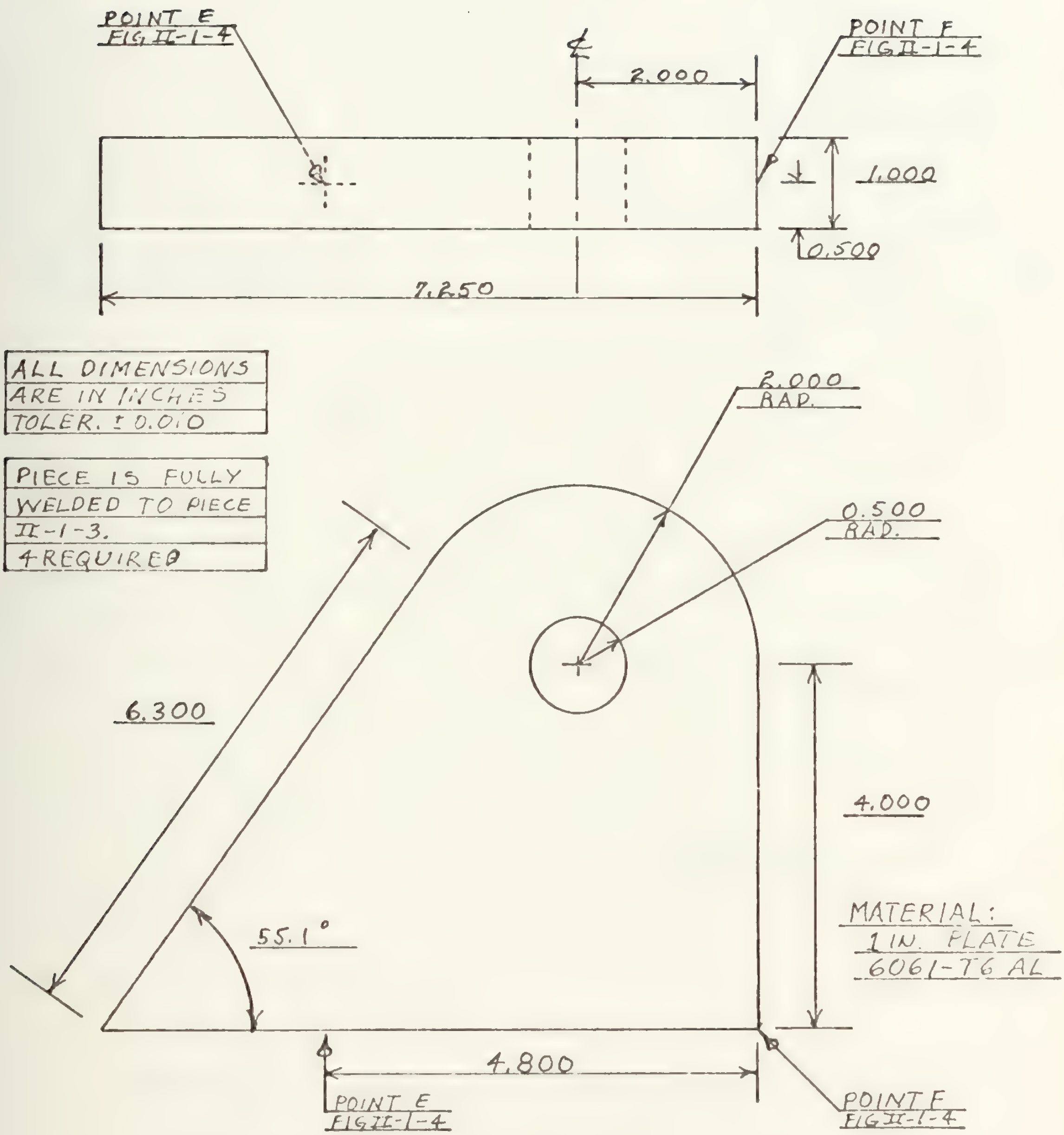


FIGURE II-1-5. PIECE II-1-5, LIFTING PAD

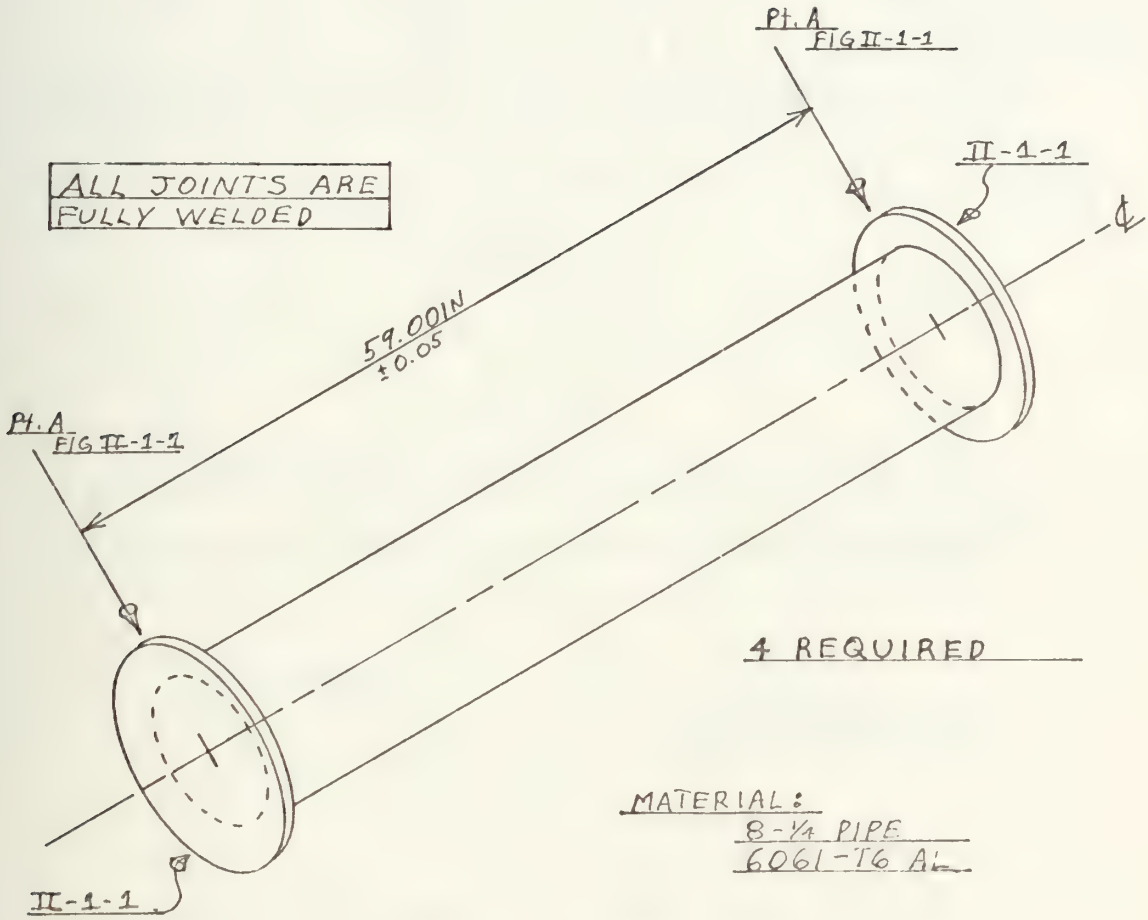
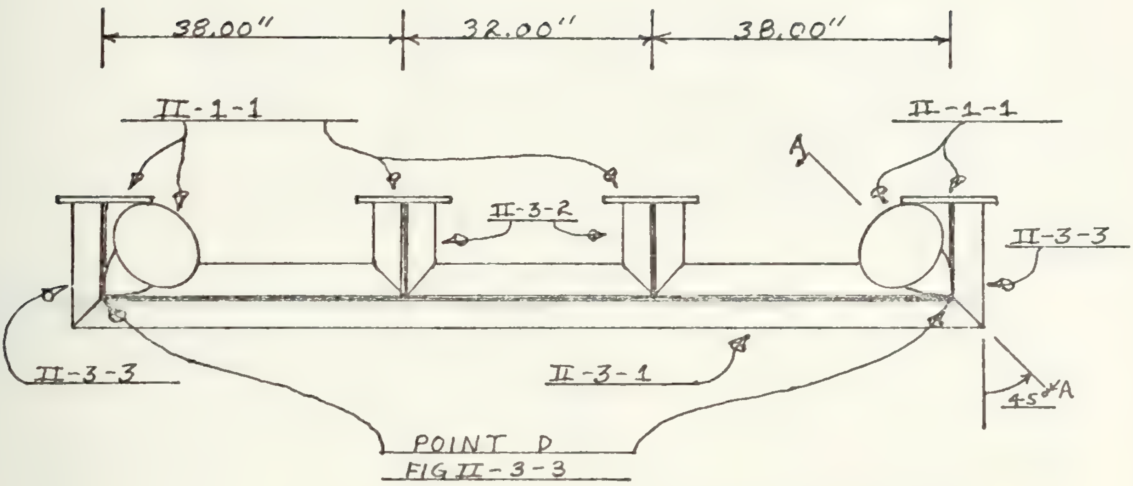
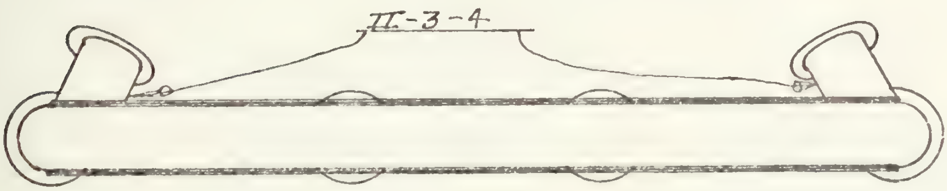


FIGURE II-2. PIECE II-2, MIDDLE SUPPORT SECTION





2 REQUIRED

ALL JOINTS ARE FULLY WELDED

FIGURE II-3. PIECE II-3, LOWER END SECTION

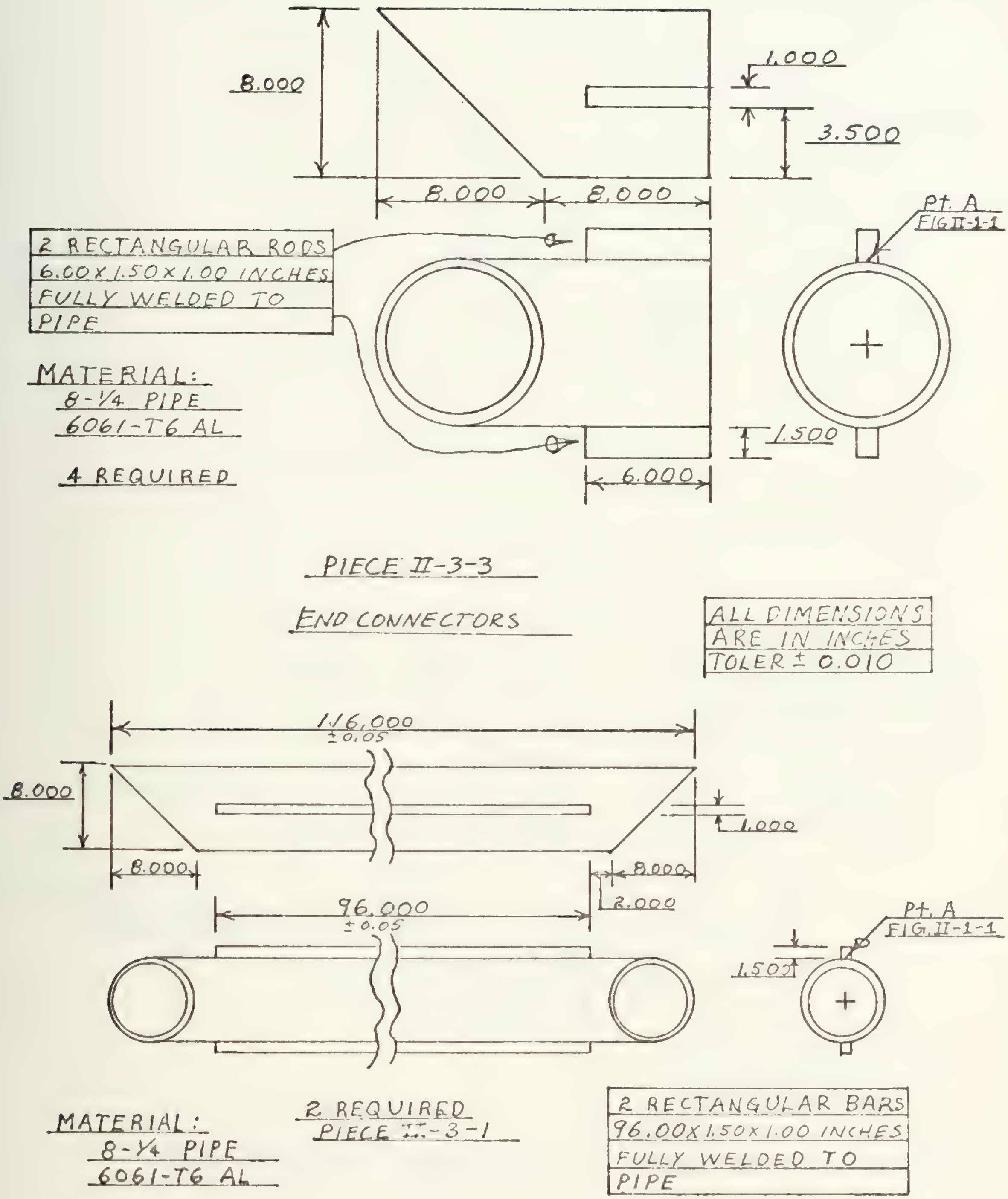
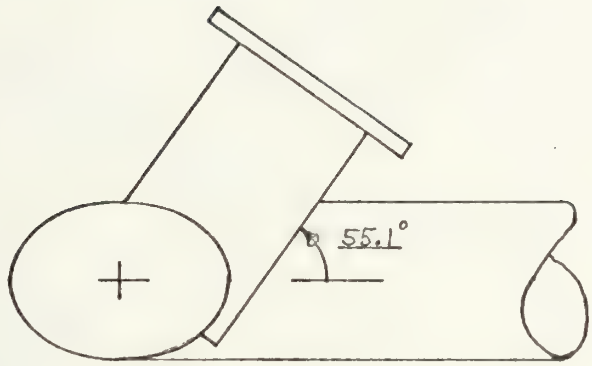


FIGURE II-3-1. PIECES II-3-1,3, SIDE CONNECTORS



LINE SECTION A-A, FIG. II-3

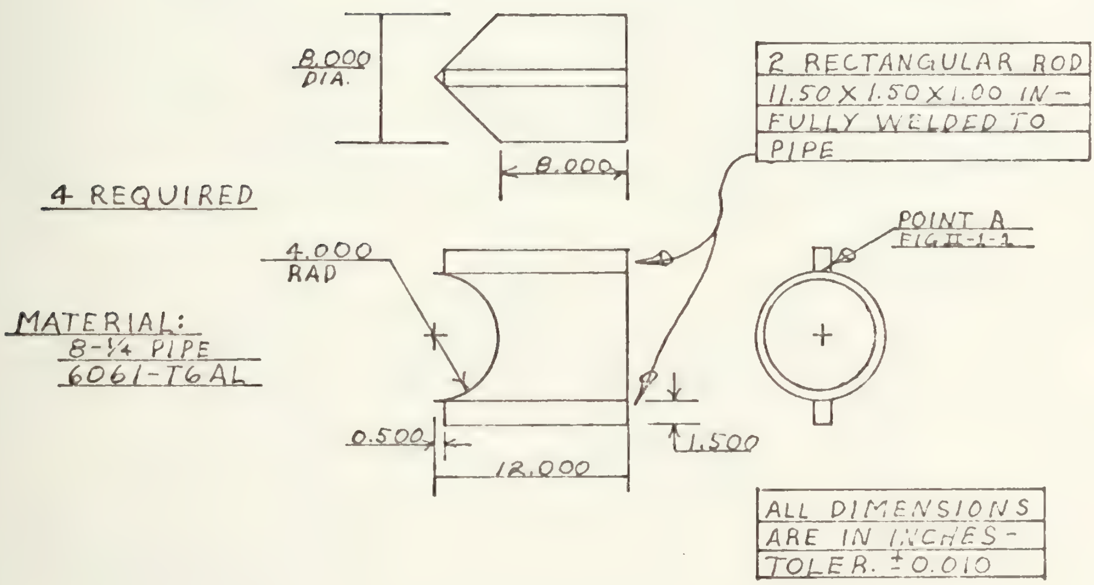


FIGURE II-3-2. PIECE II-3-2, MID-SECTION CONNECTORS

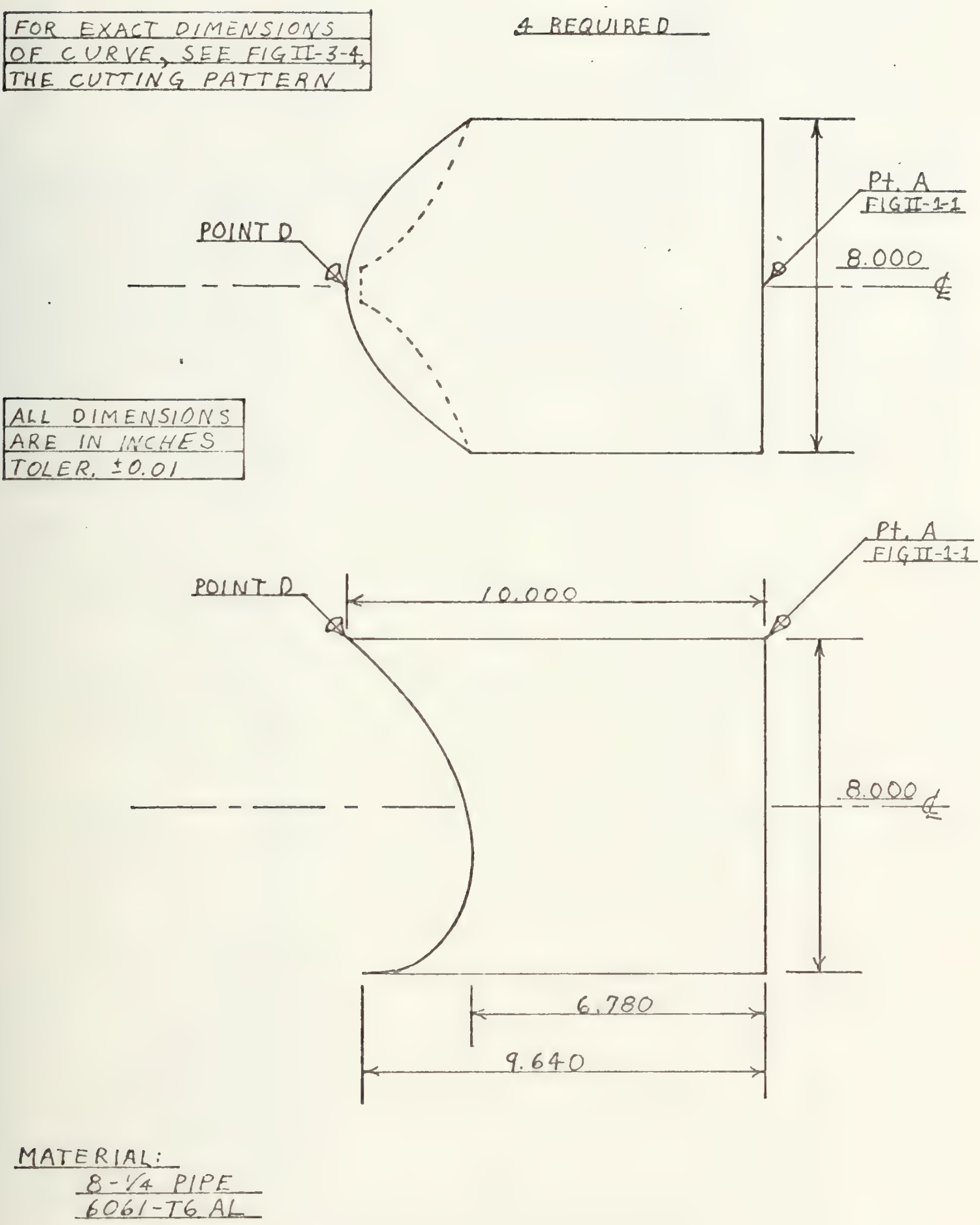


FIGURE II-3-3. PIECE II-3-4, SUPPORT CONNECTORS

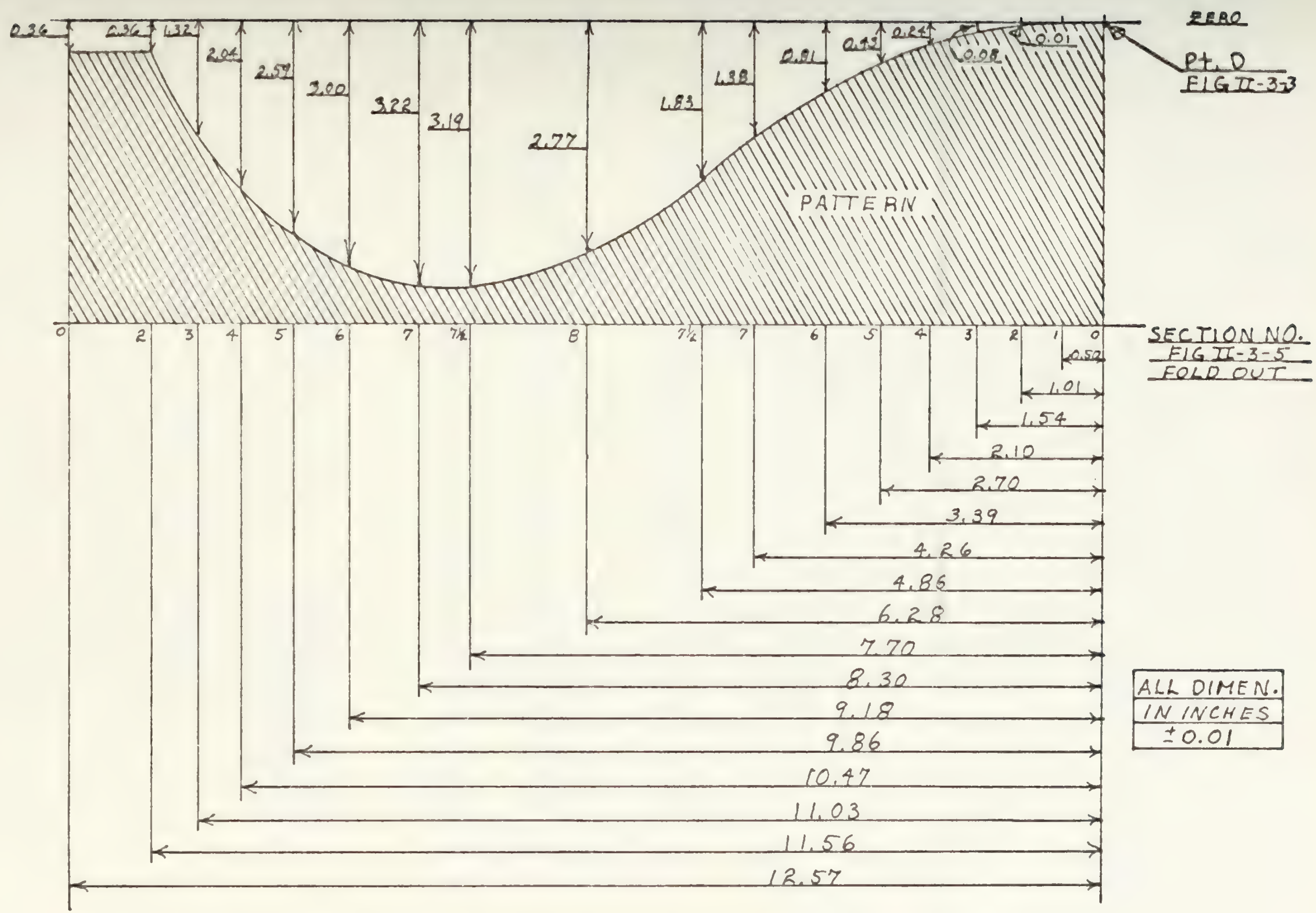


FIGURE II-3-4. 180° CIRCUMFERENTIAL SECTION OF SUPPORT CONNECTOR, CUTTING PATTERN

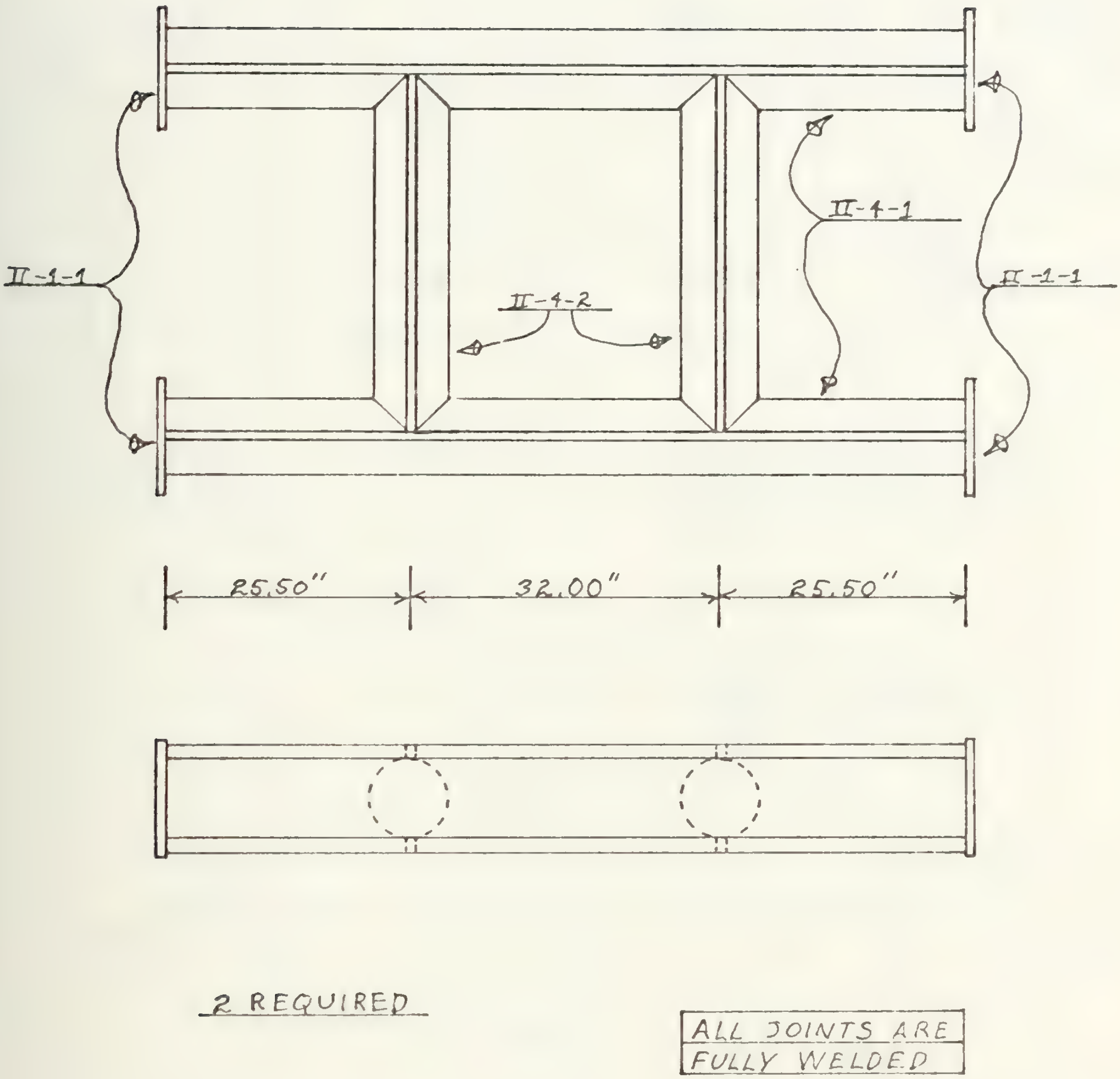


FIGURE II-4. LOWER MID-SECTION

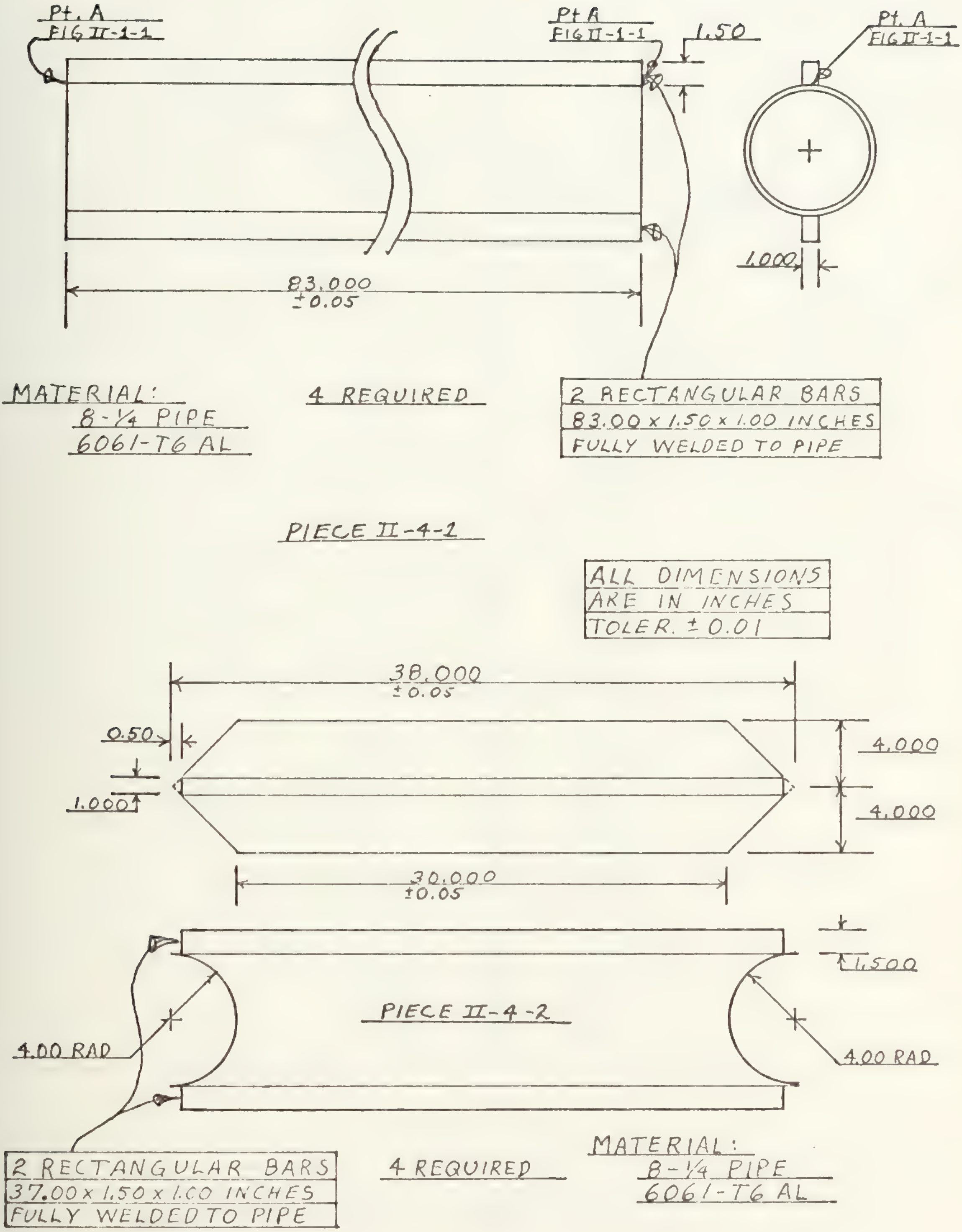


FIGURE II-4-1. PIECES II-4-1,2

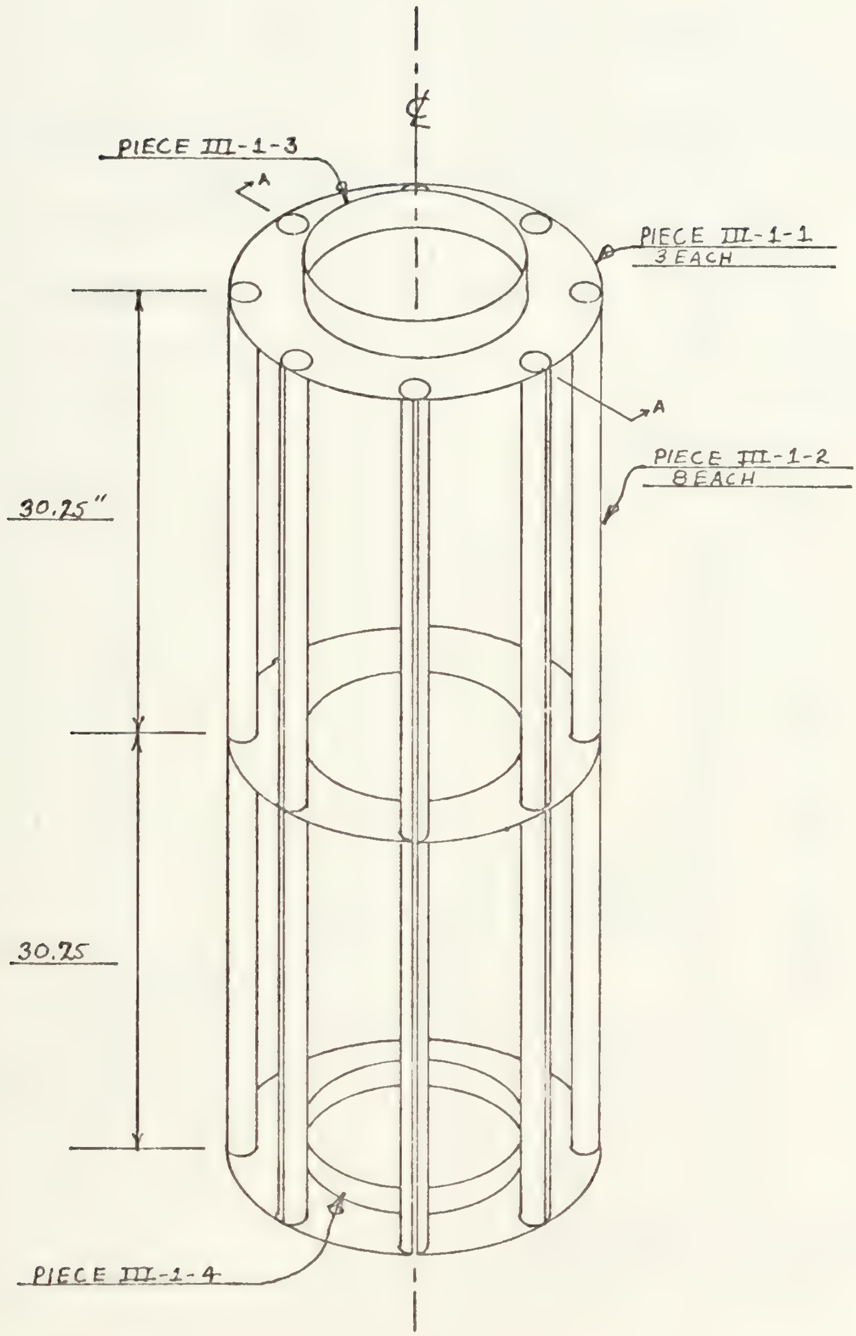


FIGURE III-1. CORING CYLINDER

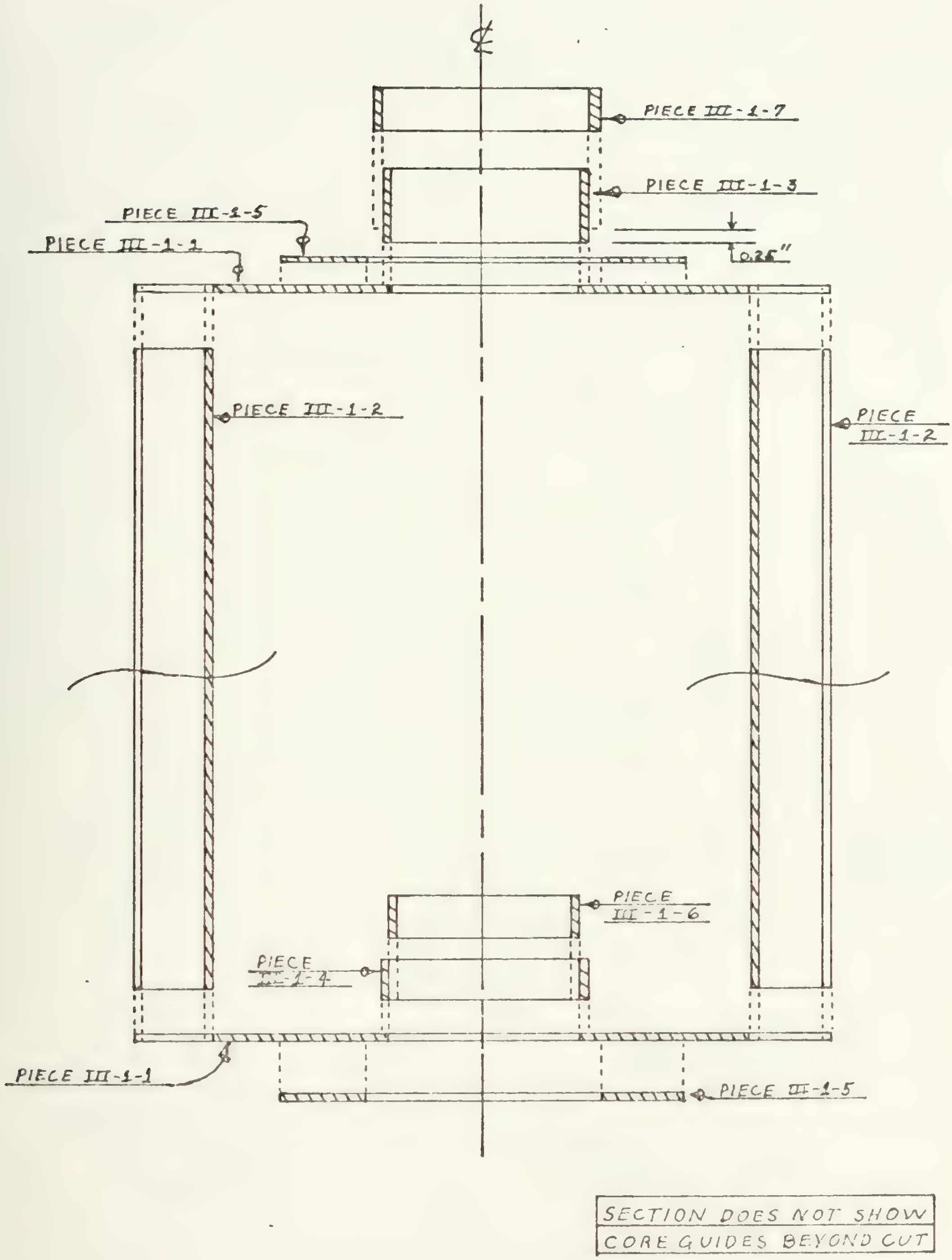


FIGURE III-1-1. SECTION A-A, SHOWING ASSEMBLY OF PIECES

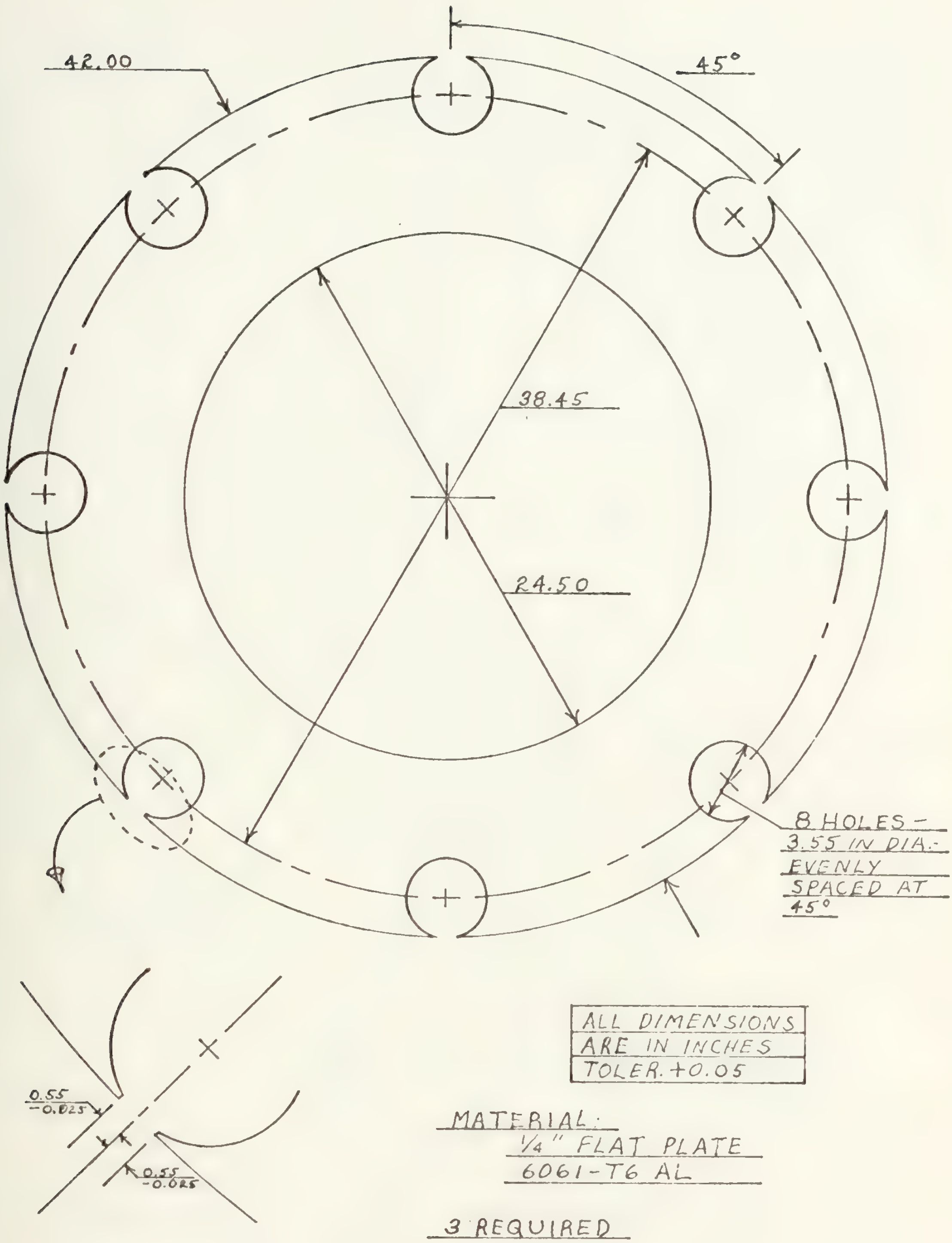


FIGURE III-1-2. PIECE III-1-1, END AND MID SECTIONS

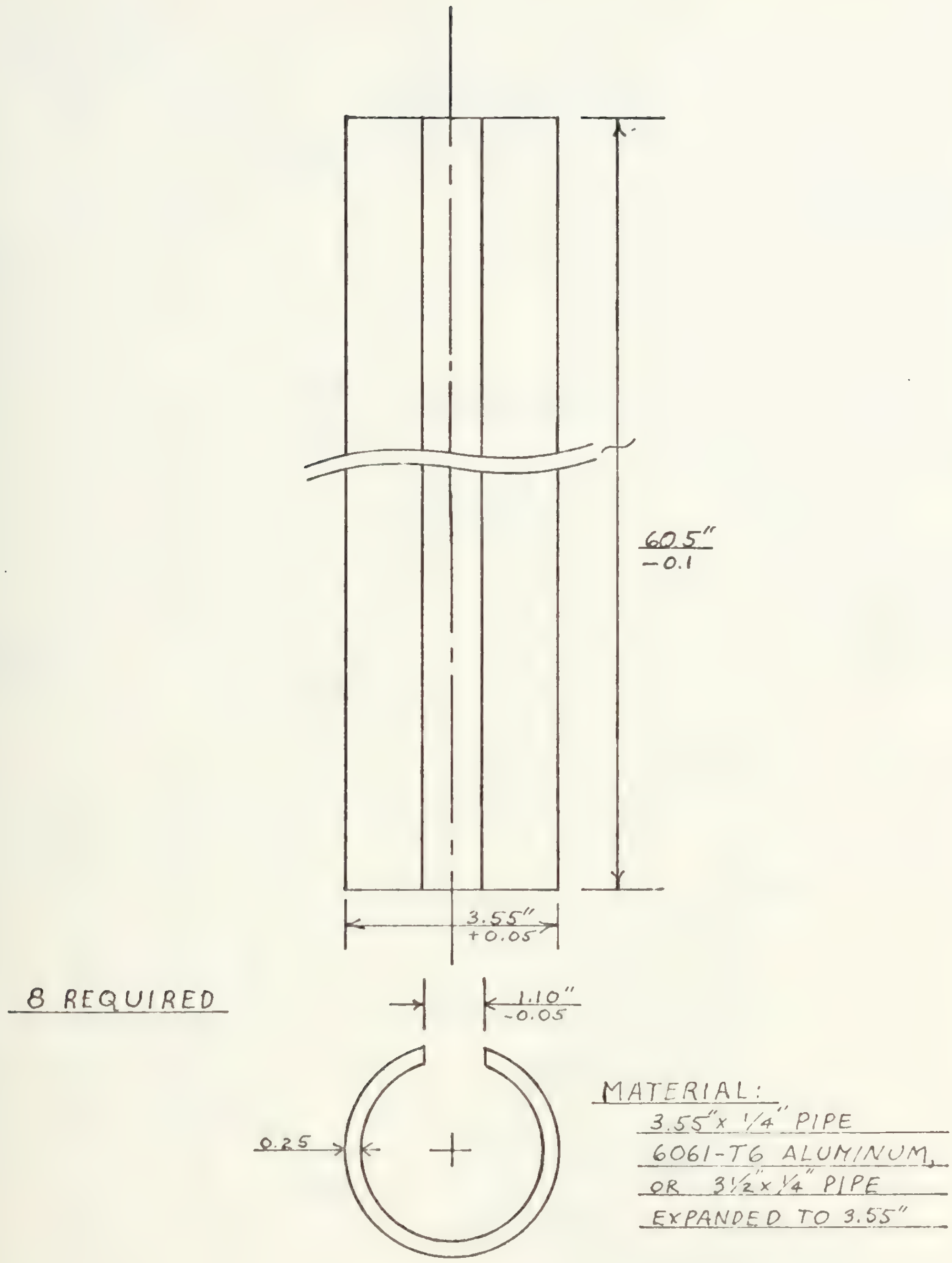


FIGURE III-1-3. PIECE III-1-2, CORE GUIDE

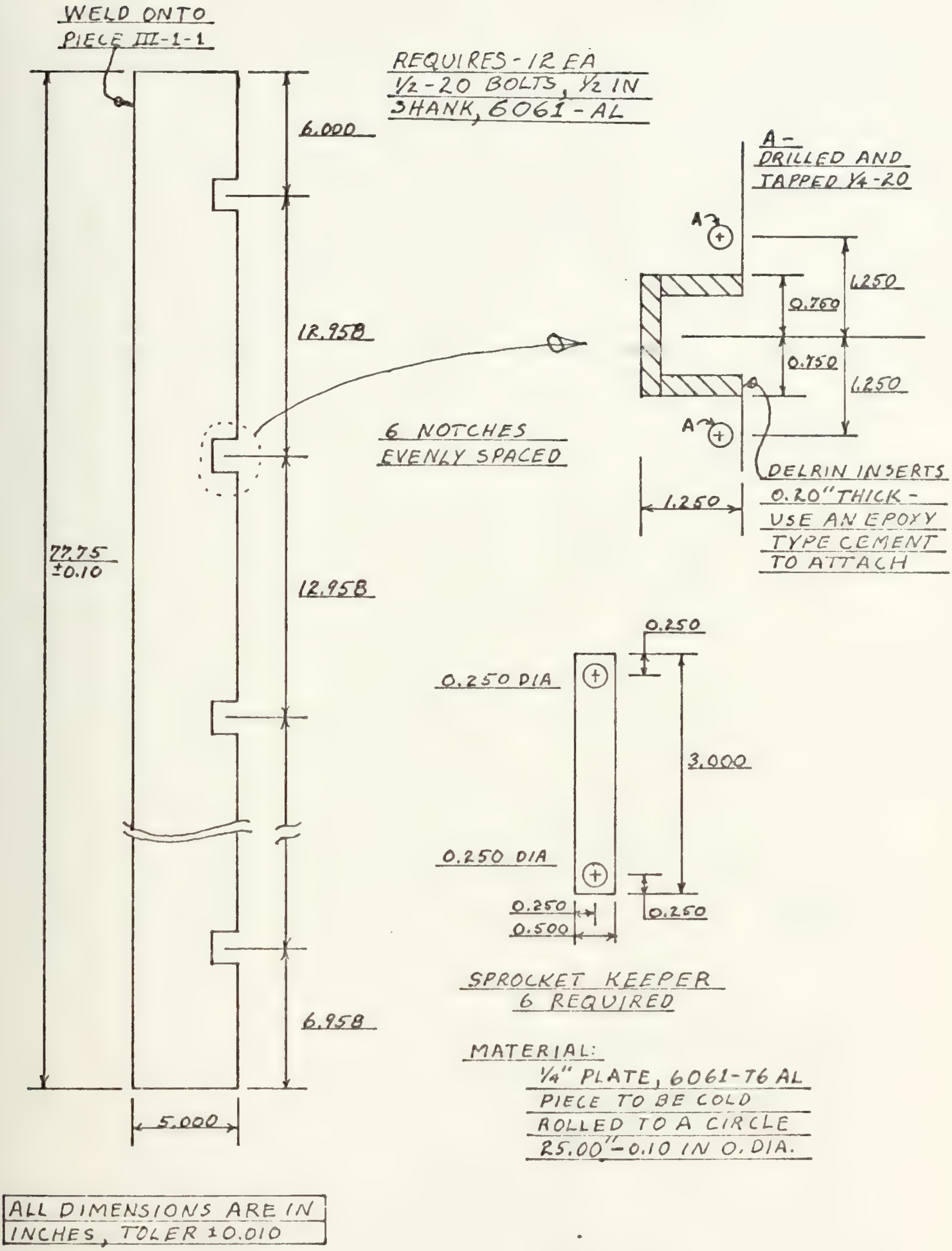


FIGURE III-1-4. PIECE III-1-3, UPPER BEARING AND DRIVE

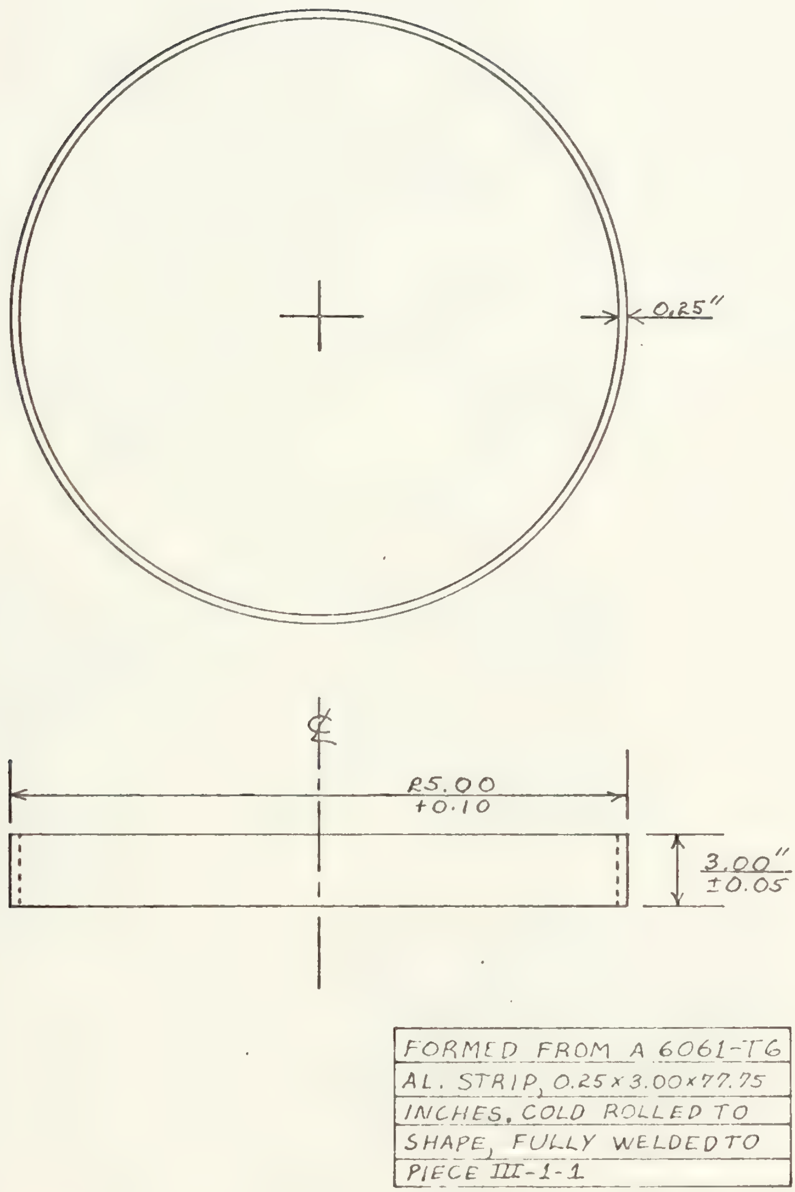


FIGURE III-1-5. PIECE III-1-4, LOWER CYLINDER BEARING SUPPORT

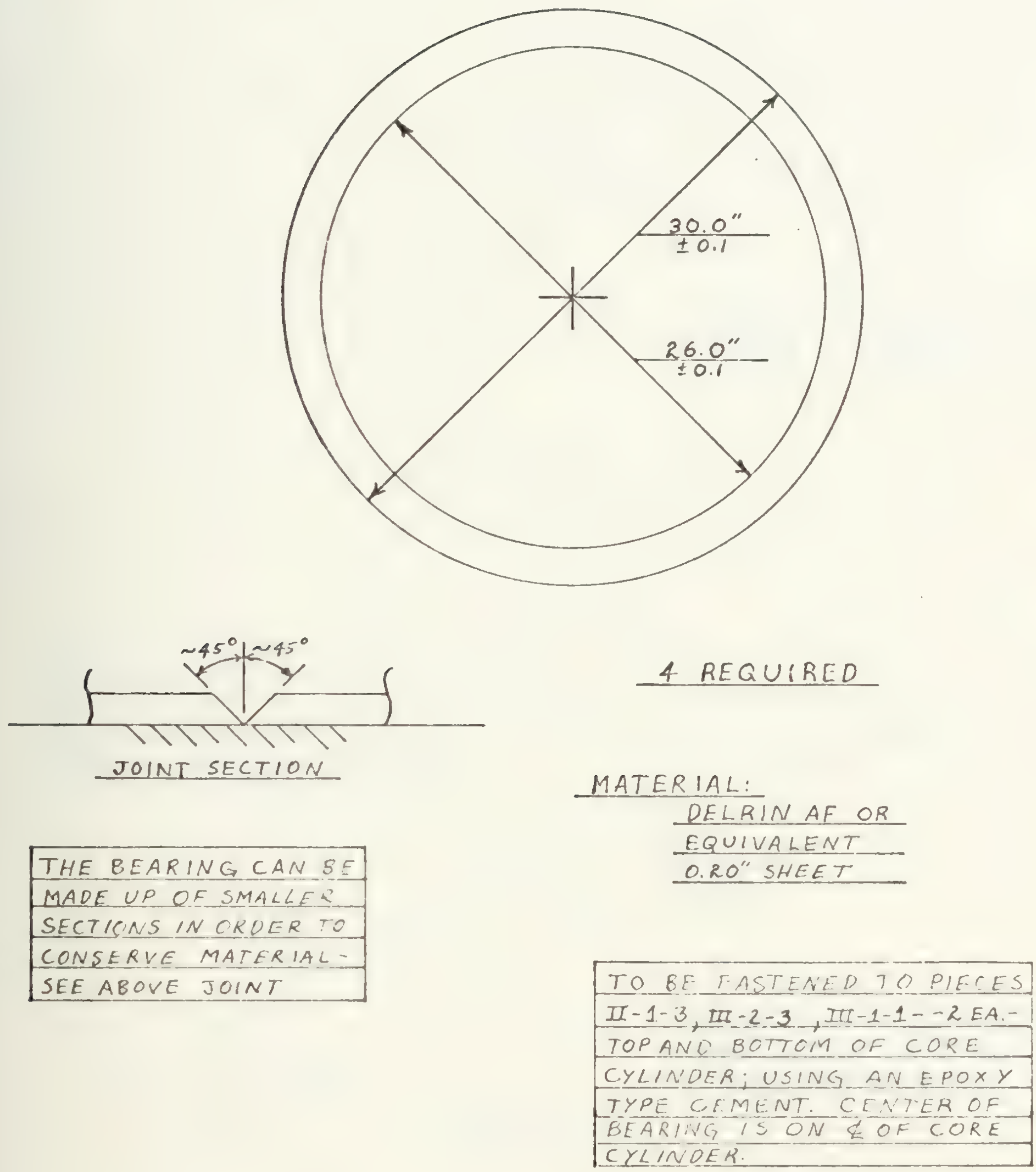


FIGURE III-1-6. PIECE III-1-5, CORING CYLINDER BEARING

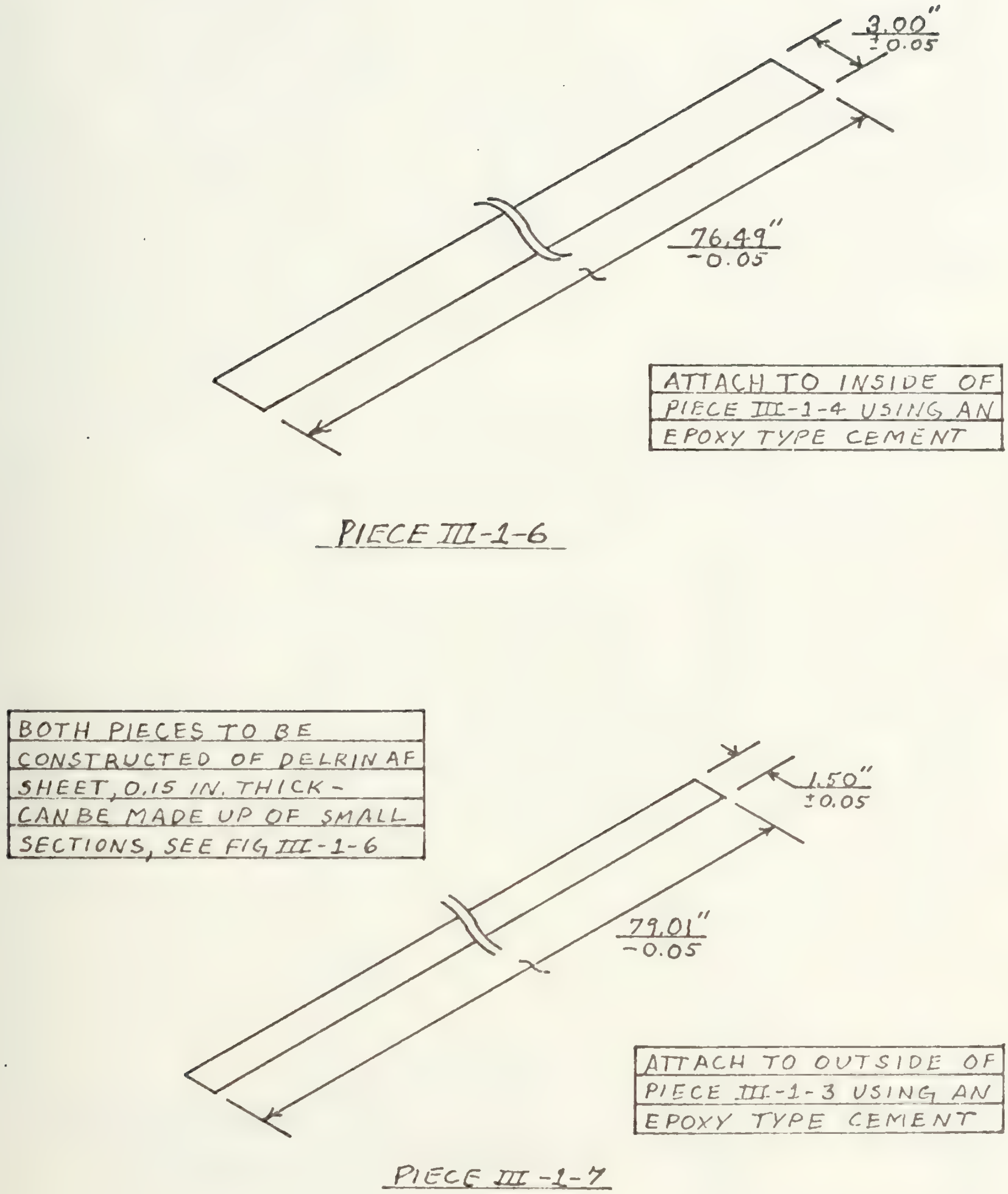


FIGURE III-1-7. PIECES III-1-6, 7,
UPPER AND LOWER CYLINDER BEARINGS

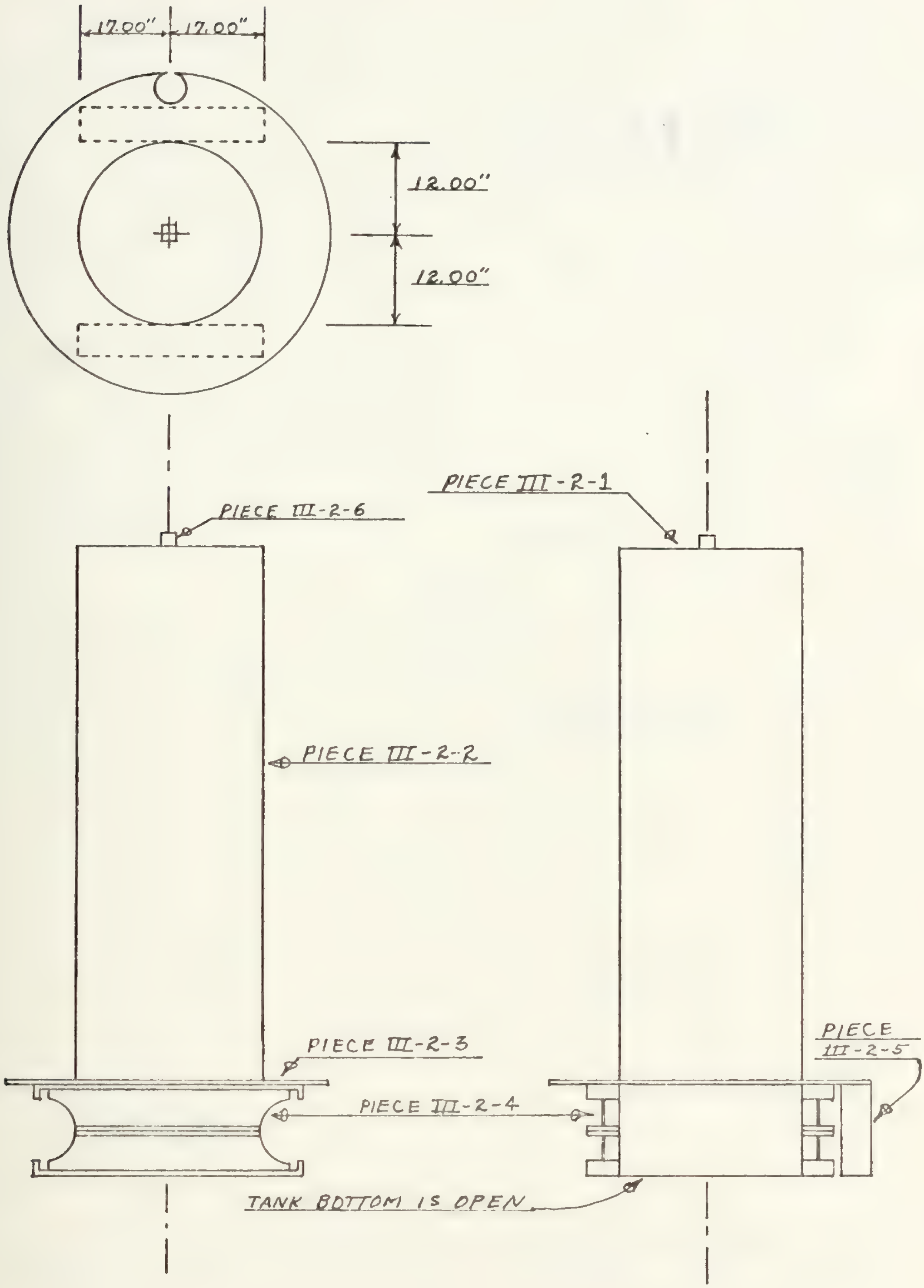


FIGURE III-2. BALLAST TANK AND CORE CYLINDER SUPPORT

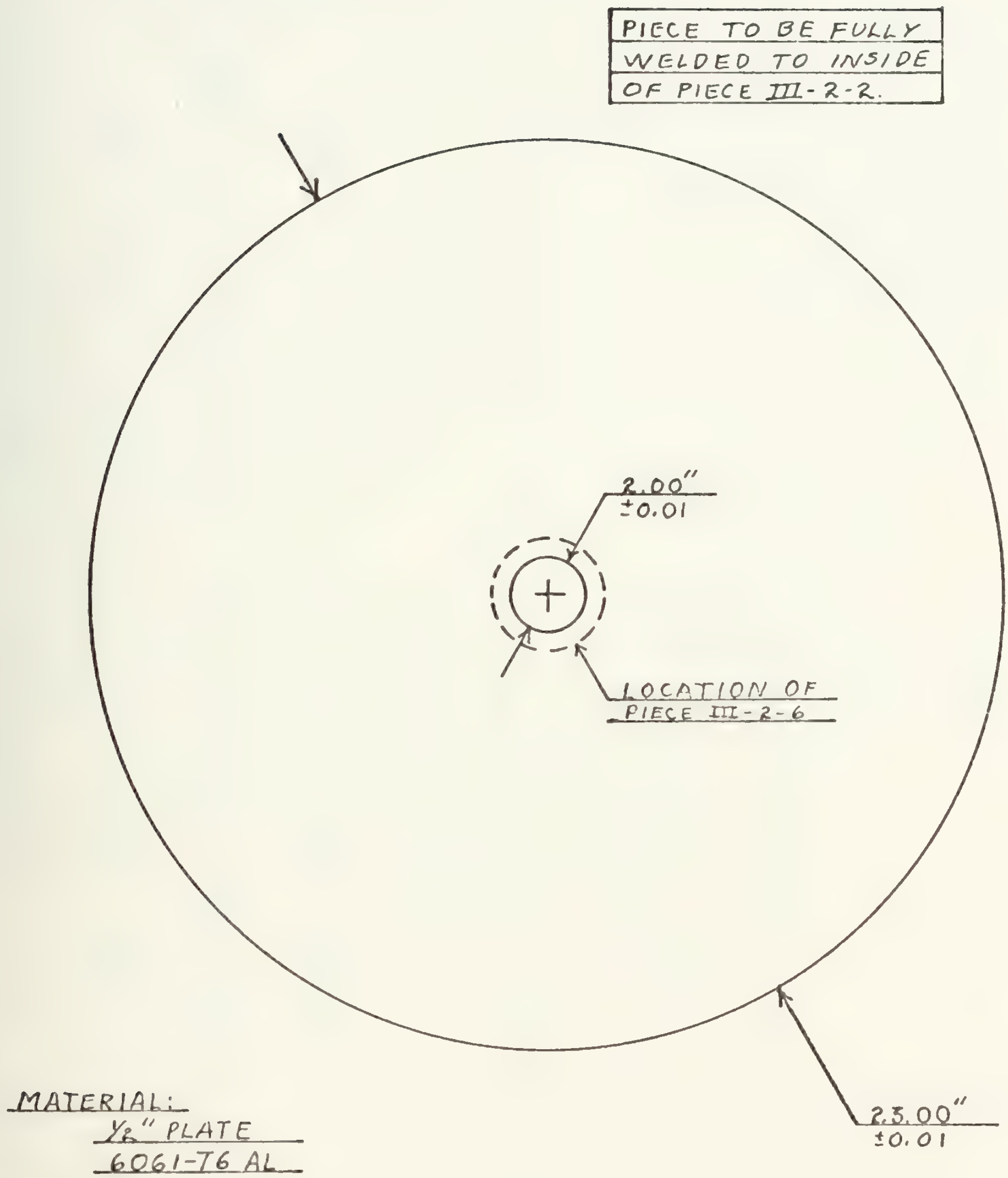


FIGURE III-2-1. PIECE III-2-1, BALLAST TANK END PLATE

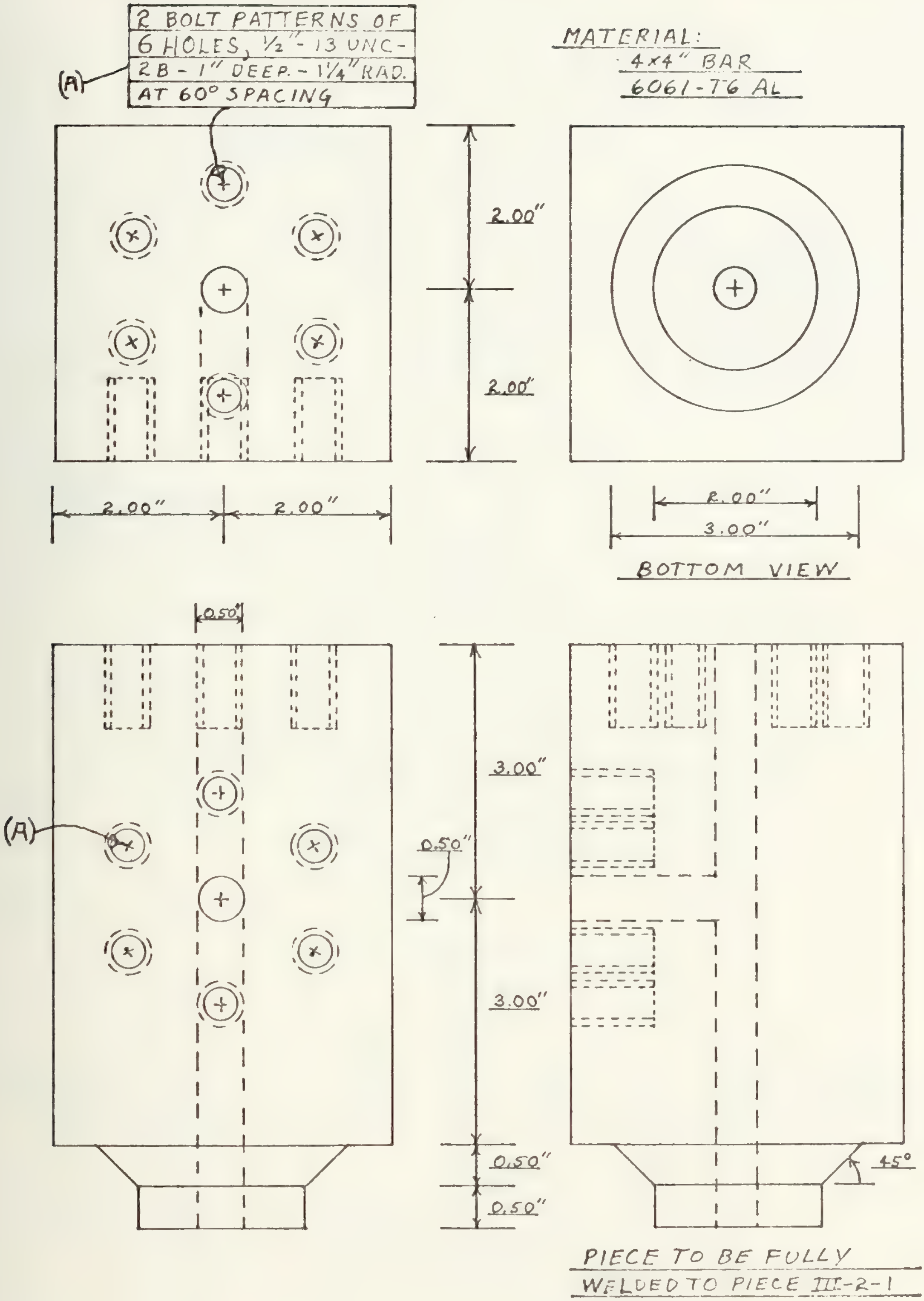
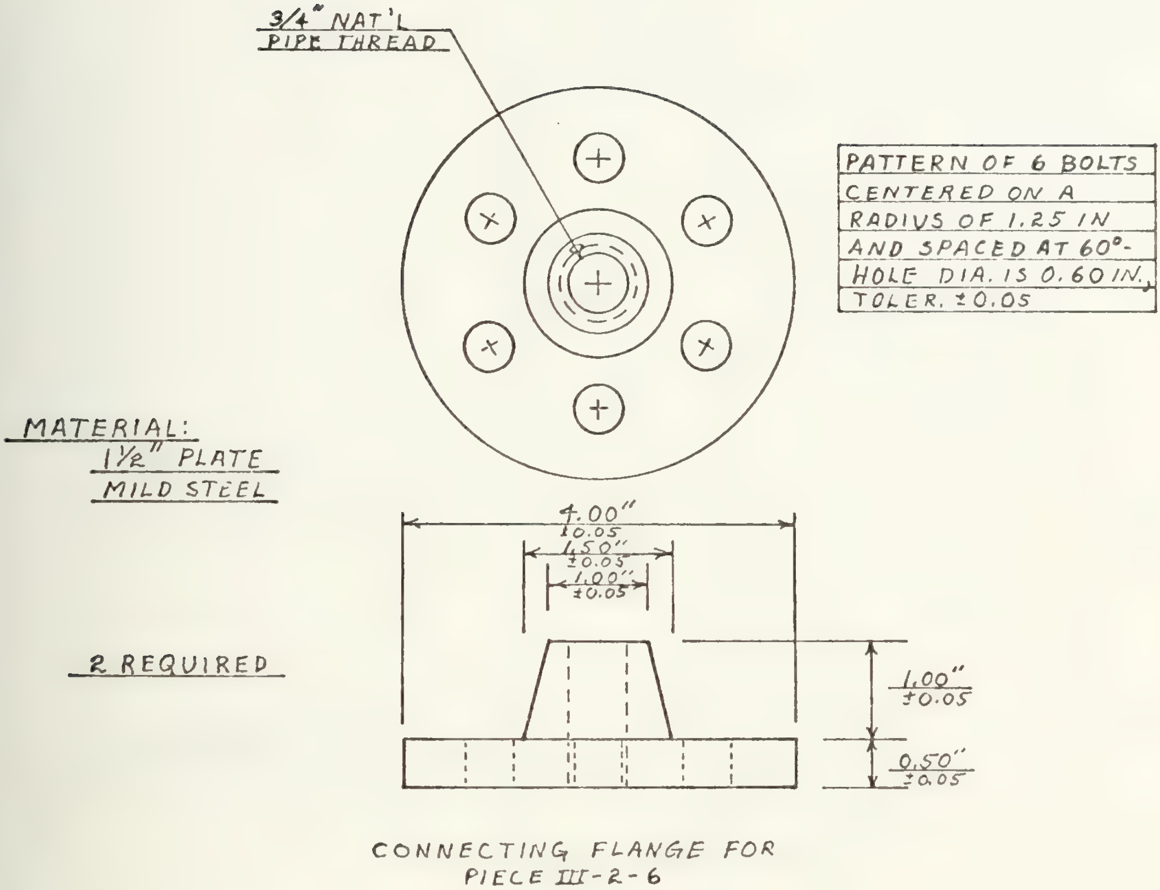


FIGURE III-2-2. PIECE III-2-6, BLOW AND VENT BLOCK



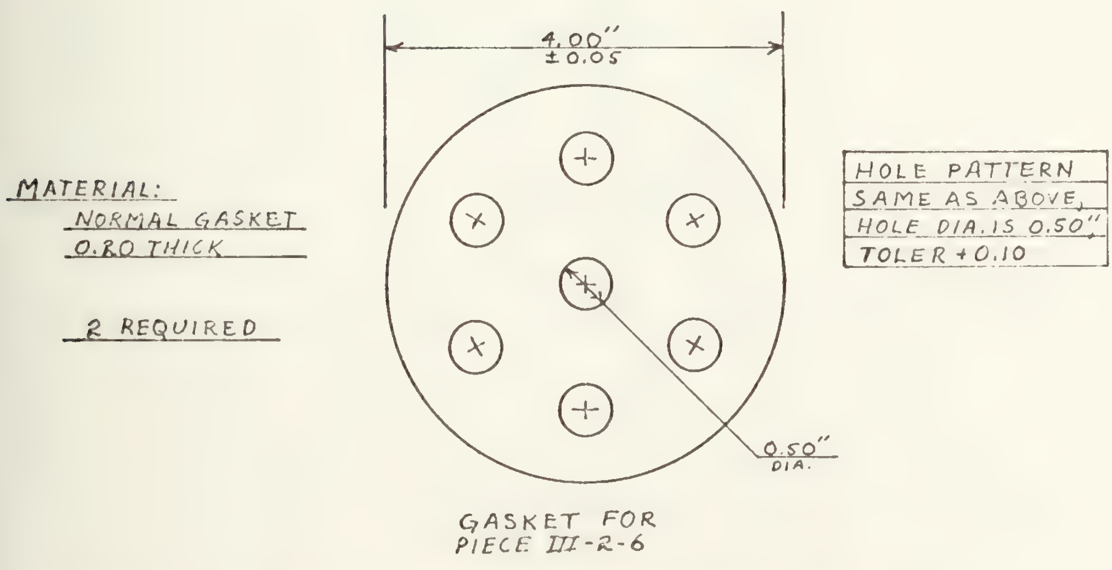


FIGURE III-2-3. CONNECTING FLANGE AND GASKET FOR PIECE III-2-6

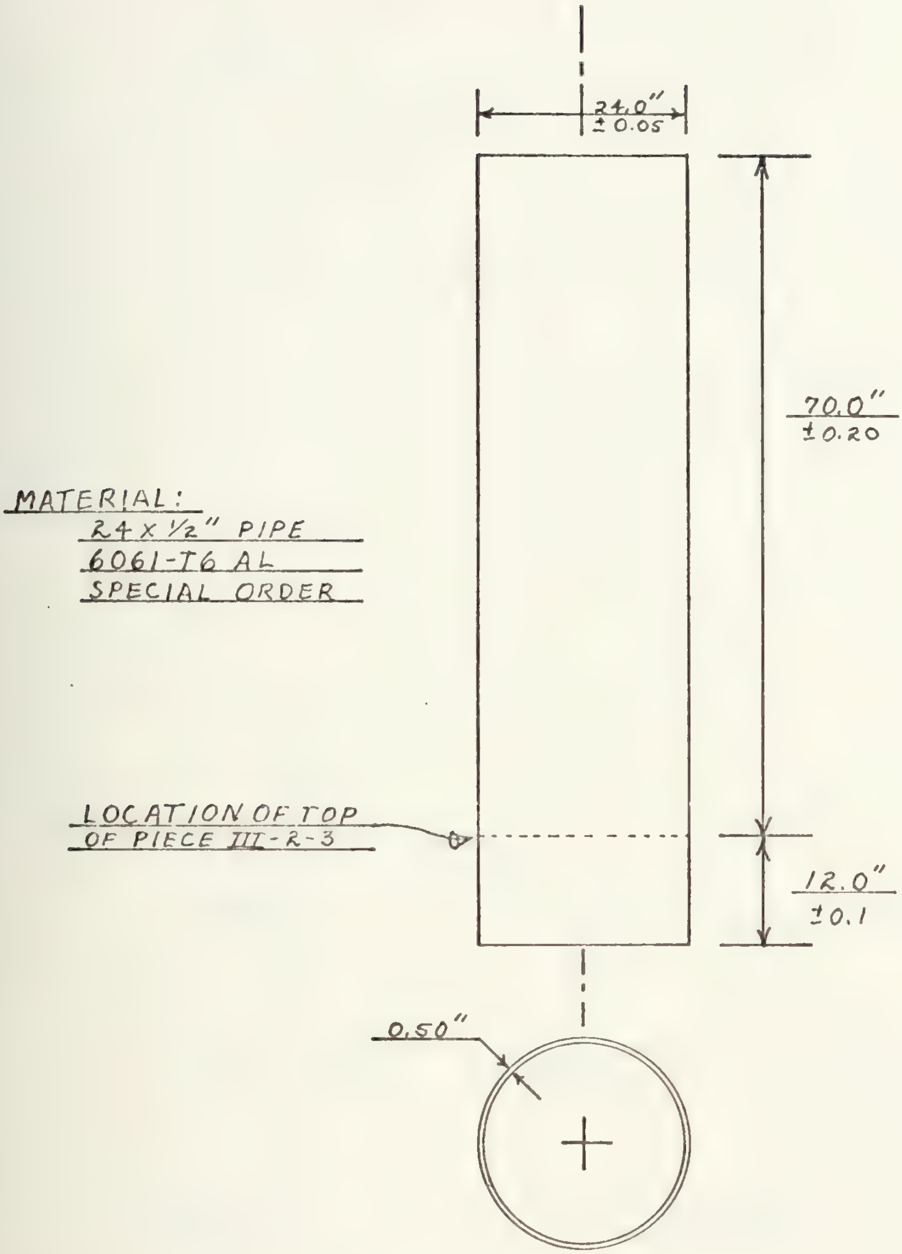


FIGURE III-2-4. PIECE III-2-2, BALLAST TANK CYLINDER

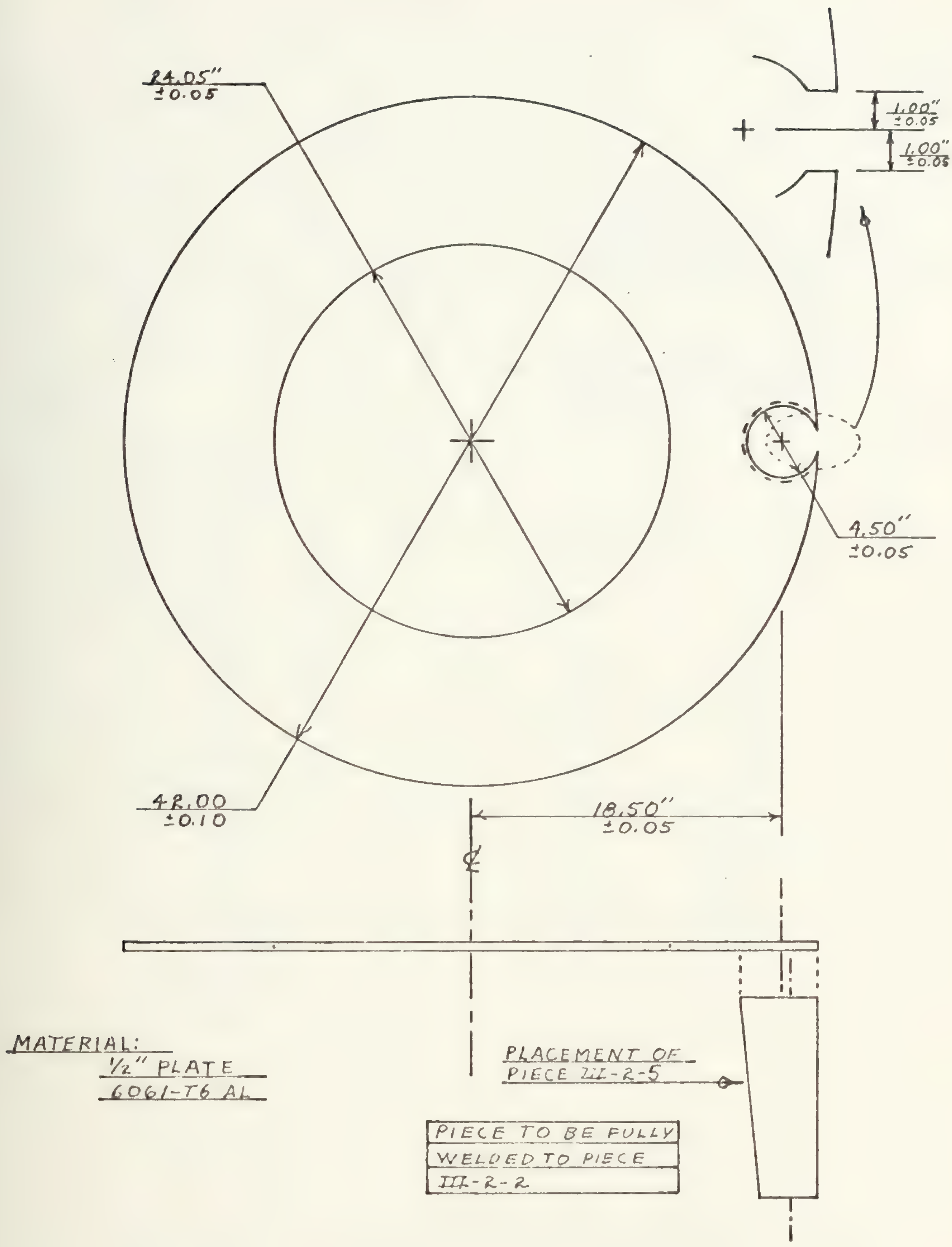


FIGURE III-2-5. PIECE III-2-3, CYLINDER SUPPORT PLATE

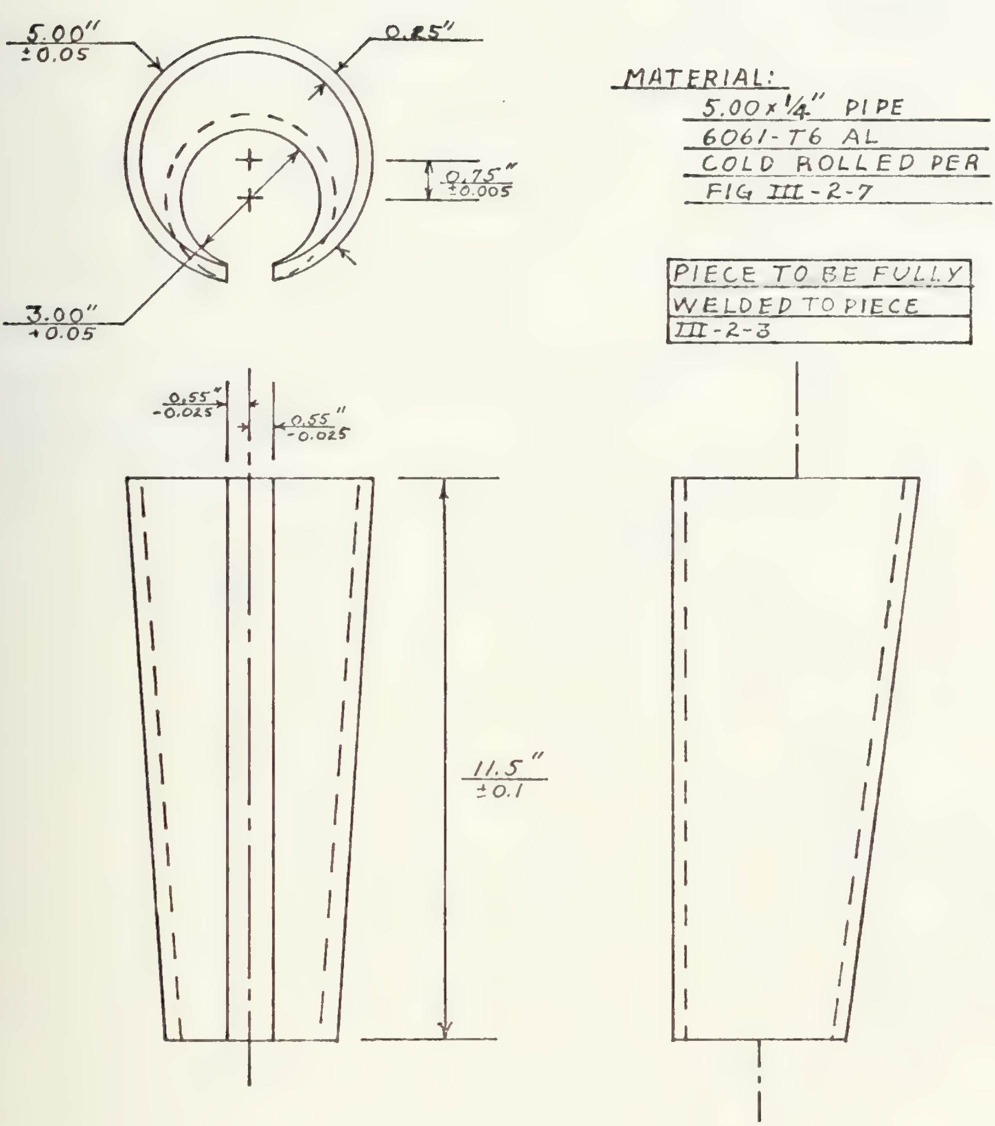


FIGURE III-2-6. PIECE III-2-5, LOWER CORE GUIDE

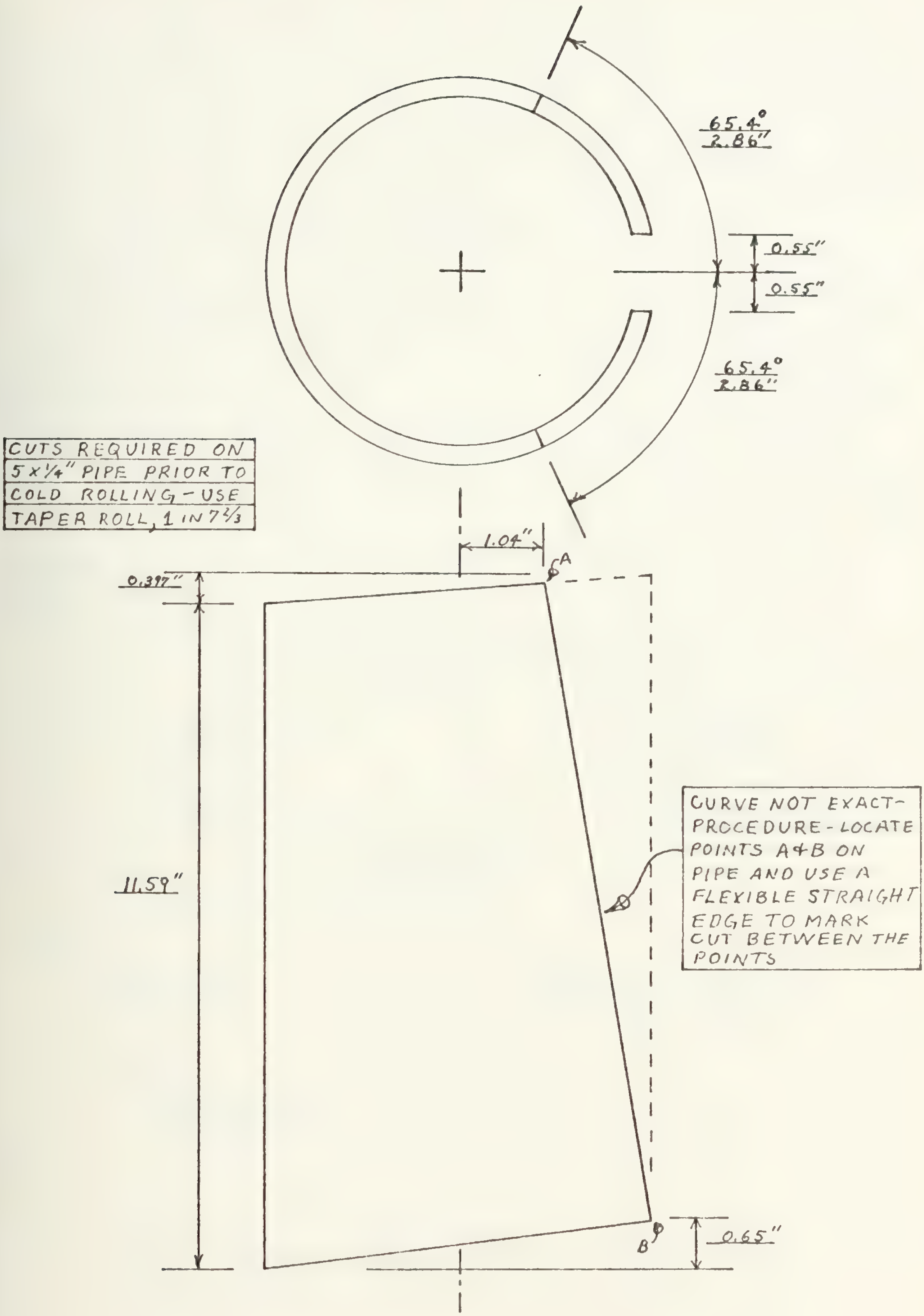


FIGURE III-2-7. BLANK FOR PIECE III-2-5

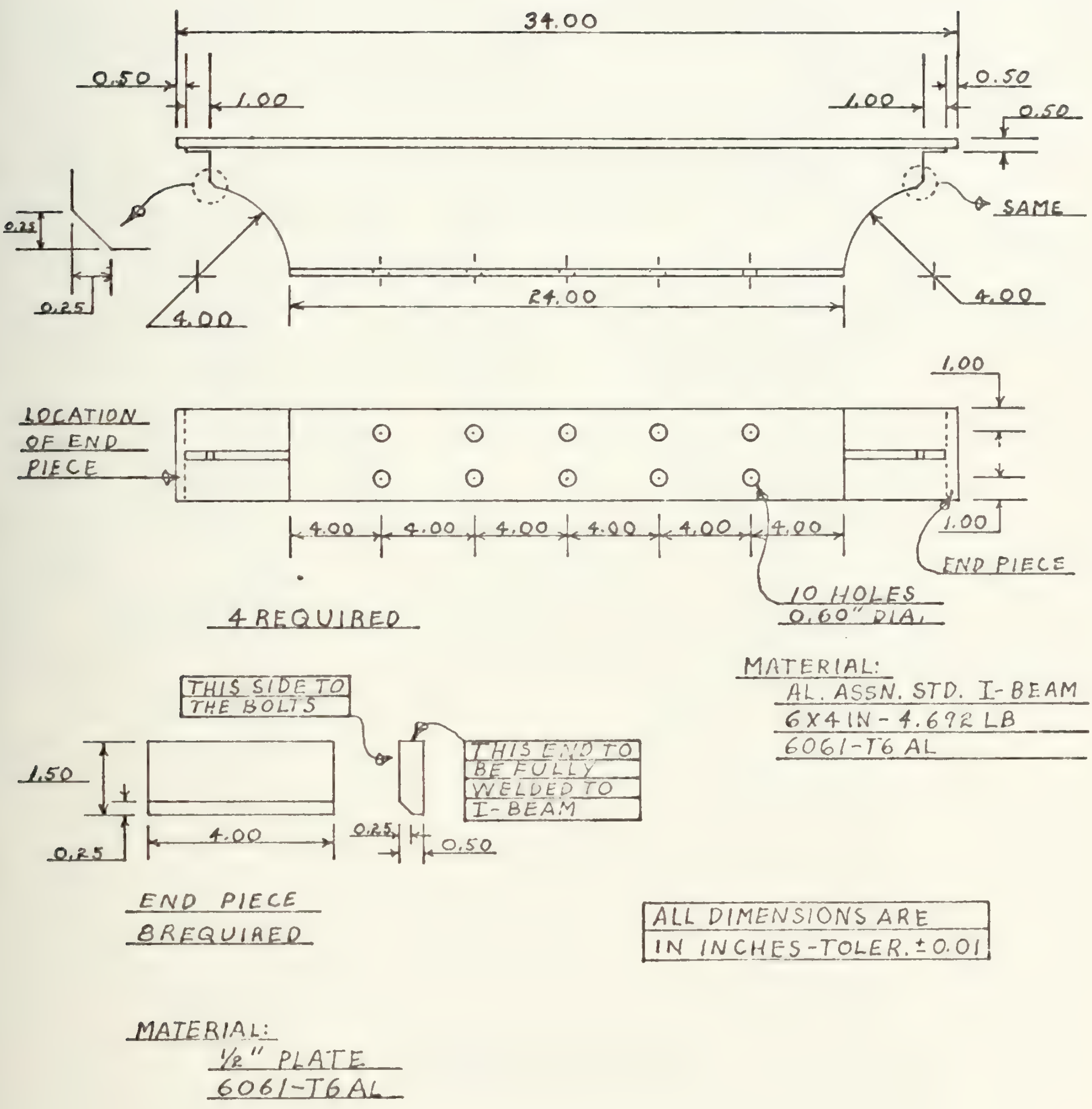


FIGURE III-2-8. PIECE III-2-4, FRAME CLAMP

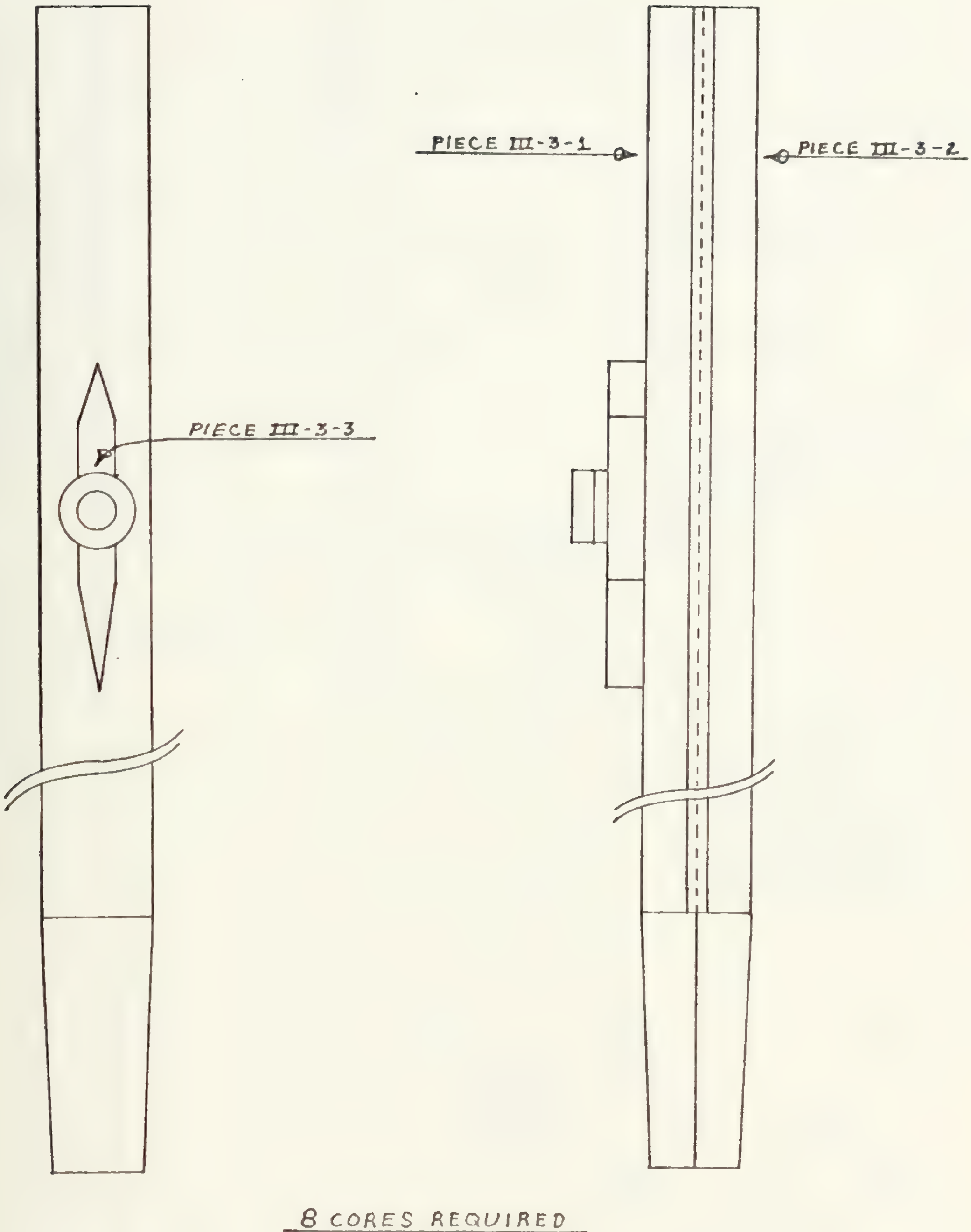


FIGURE III-3. CORE

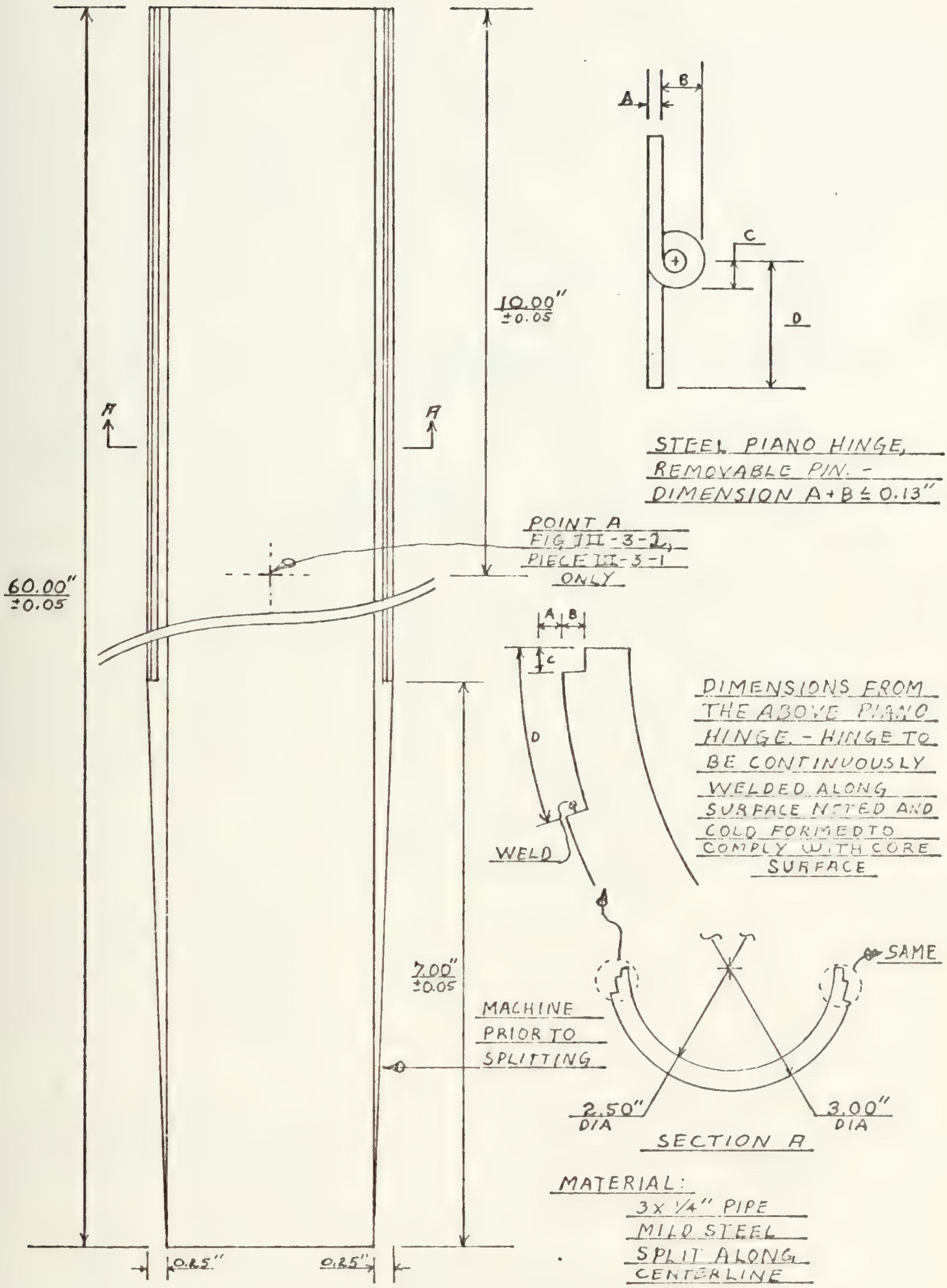


FIGURE III-3-1. PIECES III-3-1, 2, CORE BARREL

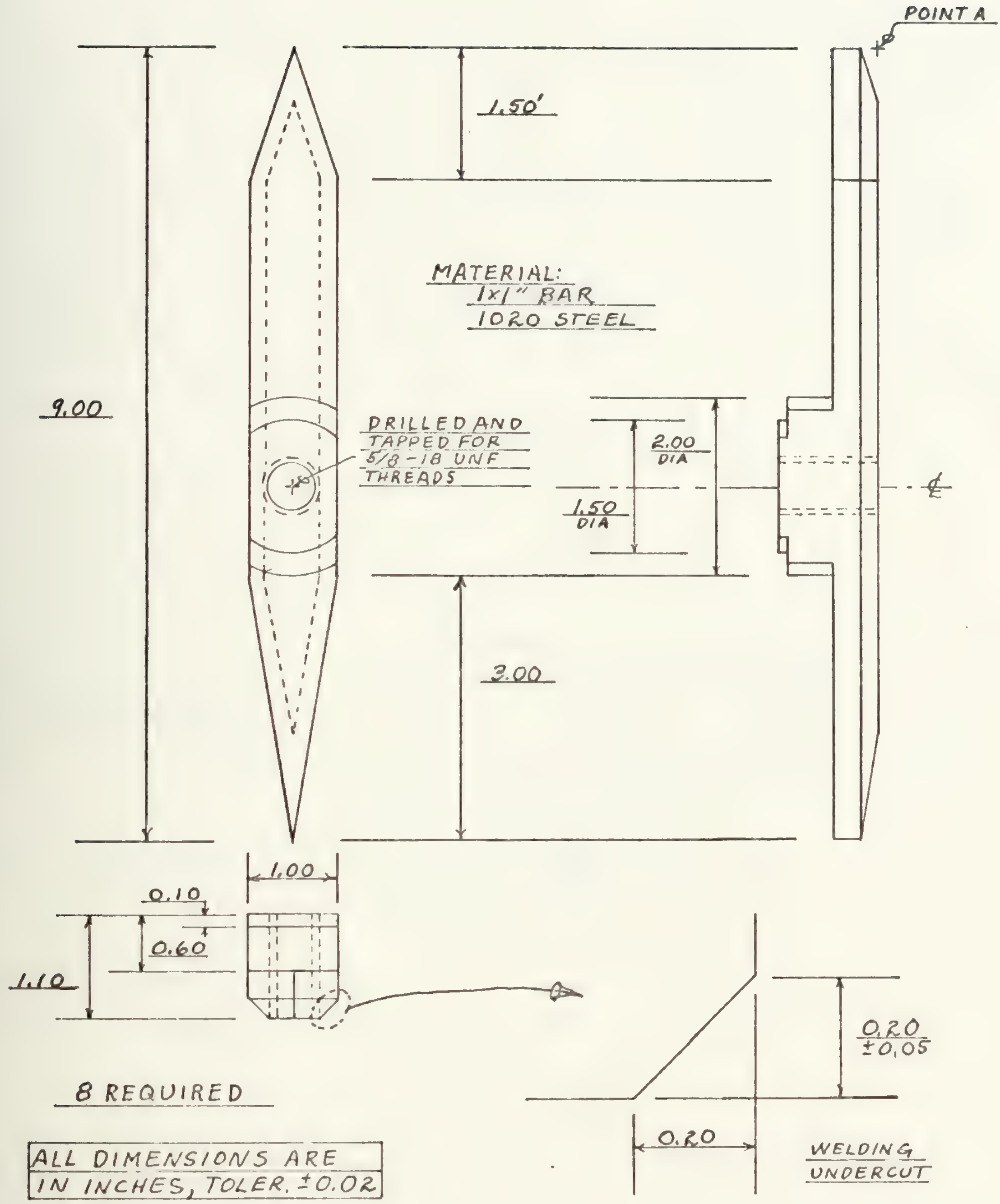


FIGURE III-3-2. PIECE III-3-3, LIFTING BEARING BLOCK

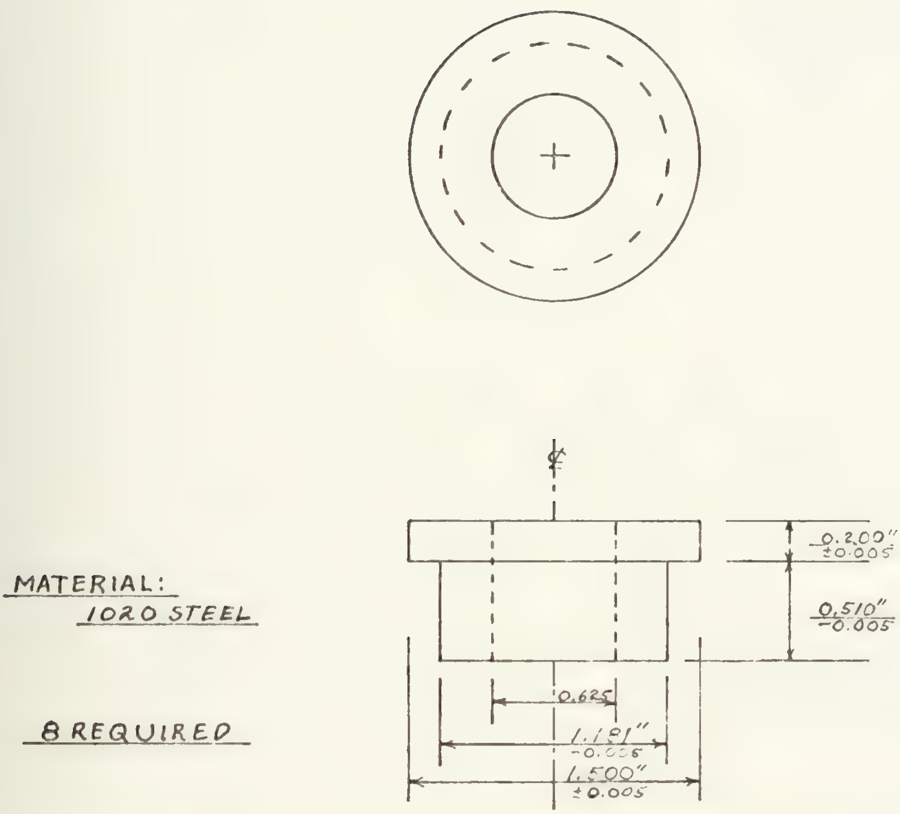


FIGURE III-3-3. PIECE III-3-4, BEARING RETAINER

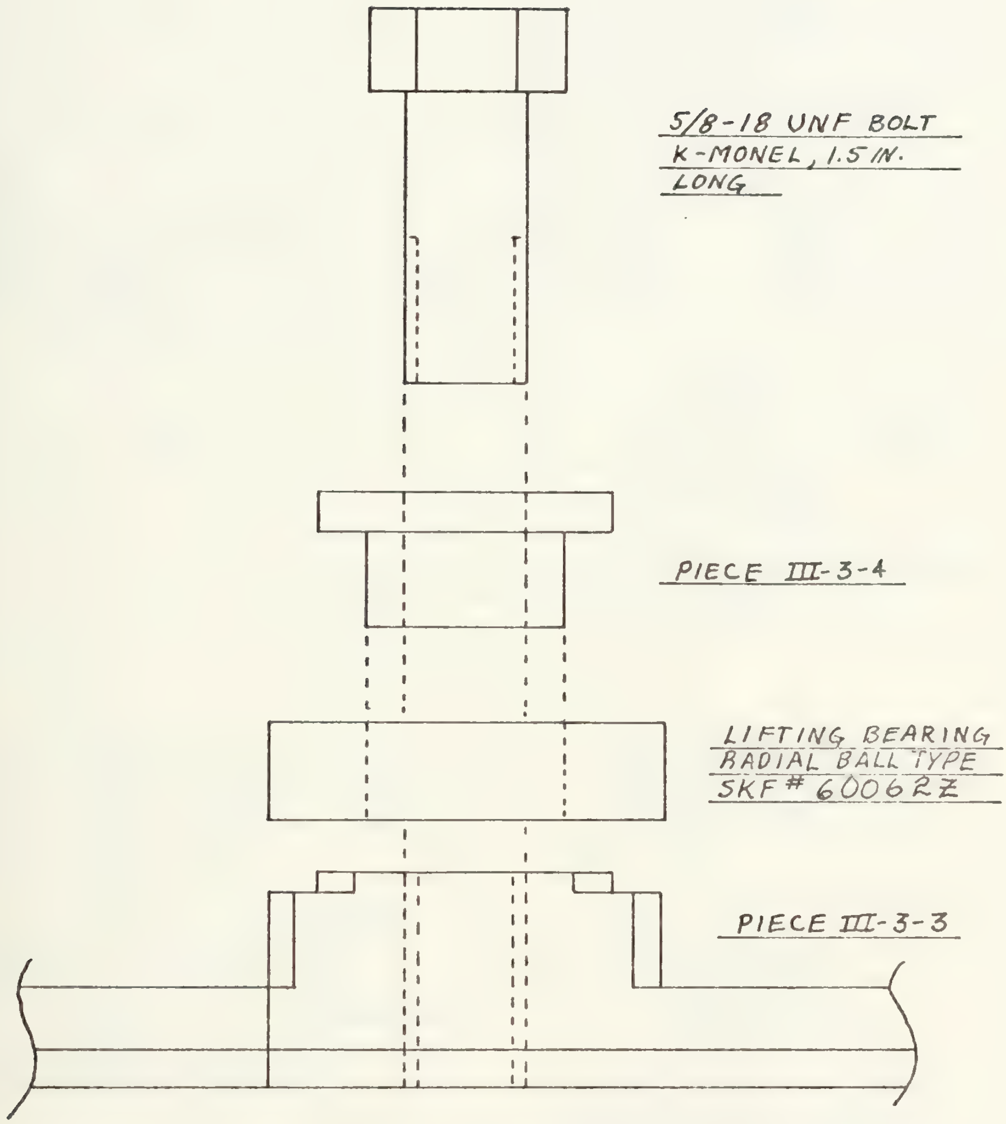


FIGURE III-3-4. ASSEMBLY OF LIFTING BEARING AND BLOCK

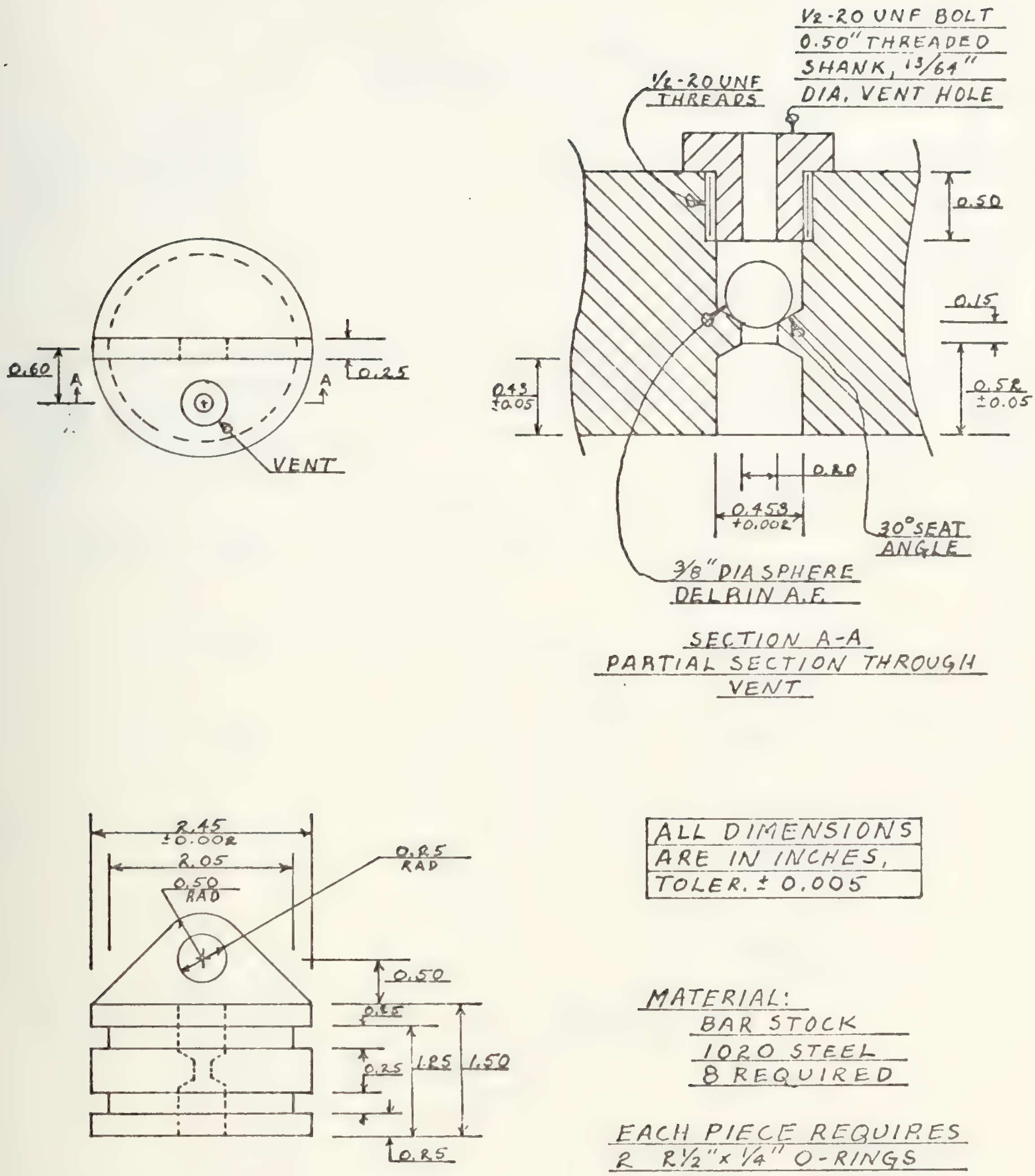


FIGURE III-3-5. PIECE III-3-5, CORE PISTON

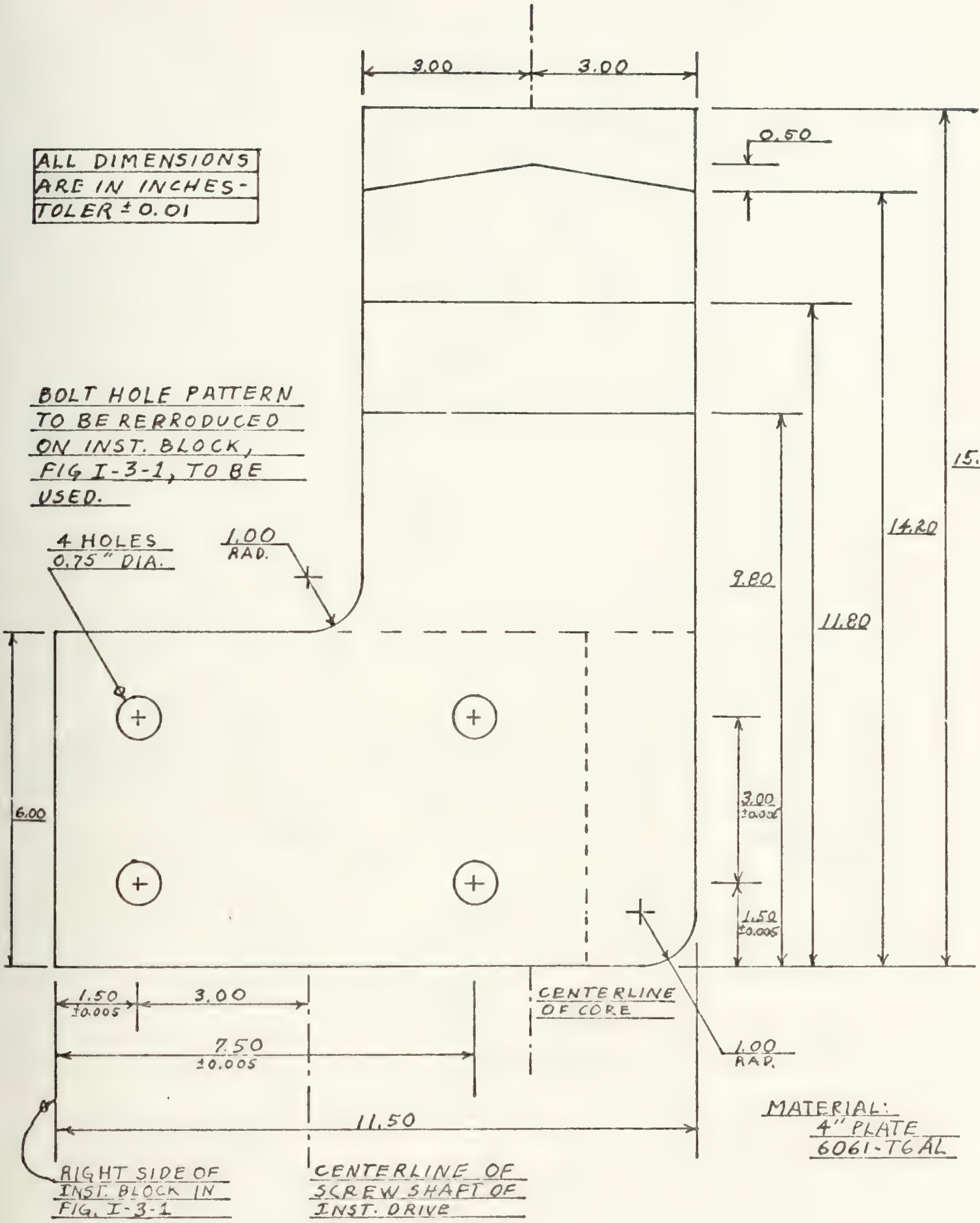


FIGURE III-4. PIECE III-4-1, INSTRUMENT DRIVE - CORE CONNECTOR, VIEW FROM CORE

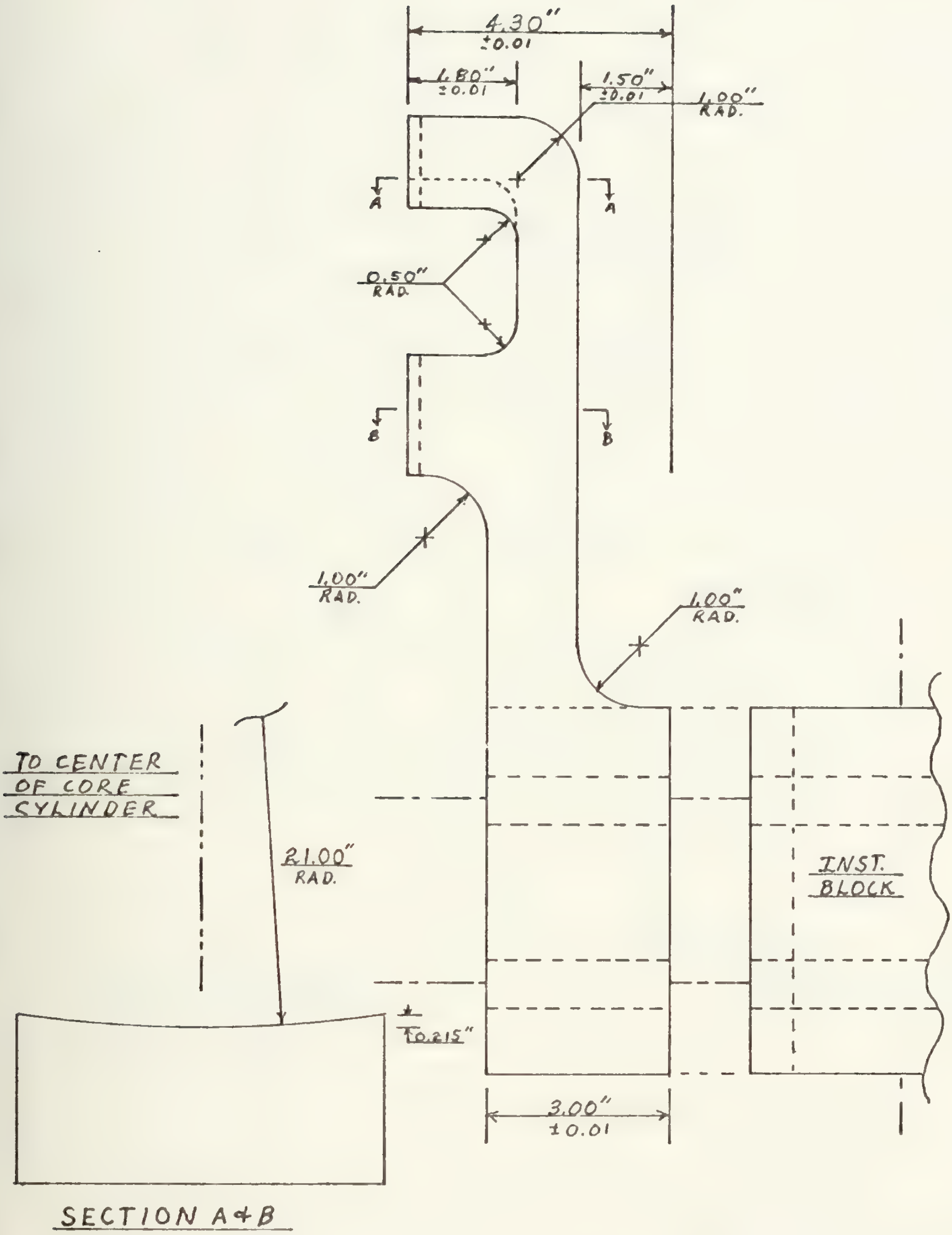


FIGURE III-4-1. PIECE III-4-1, SIDE VIEW

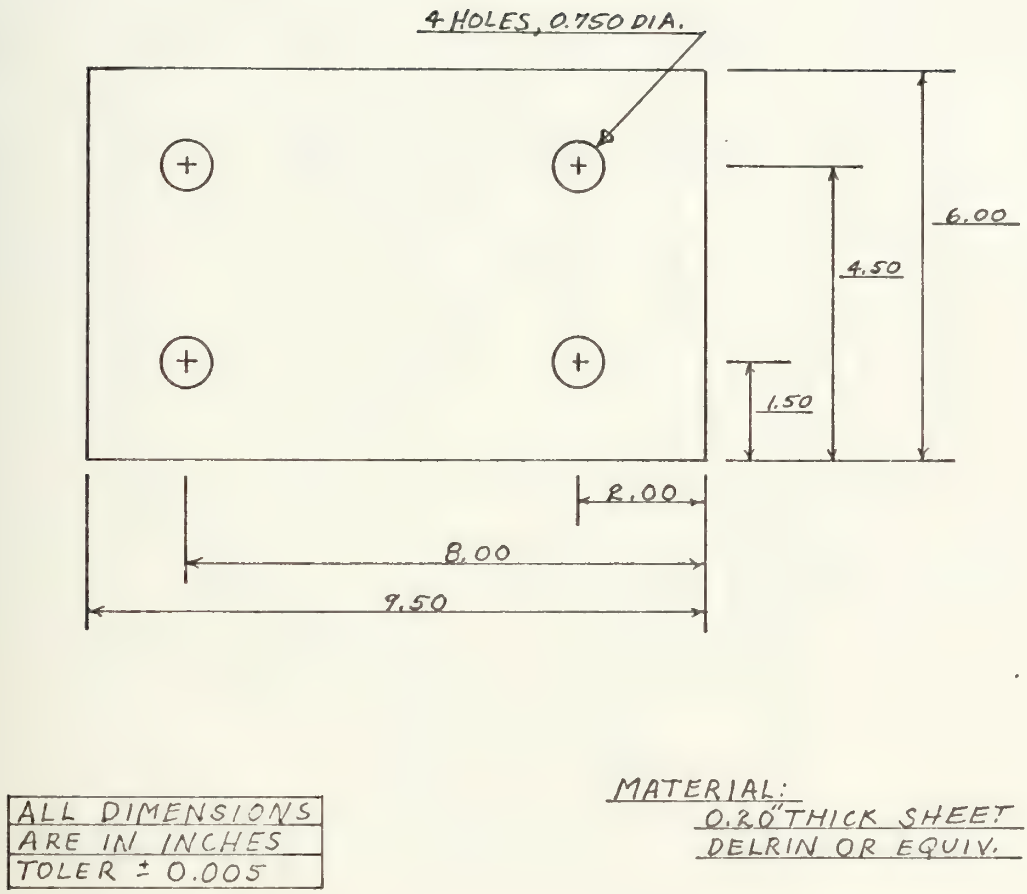


FIGURE III-4-2. PIECE III-4-2, INSTRUMENT DRIVE - CORE CONNECTOR INSULATOR

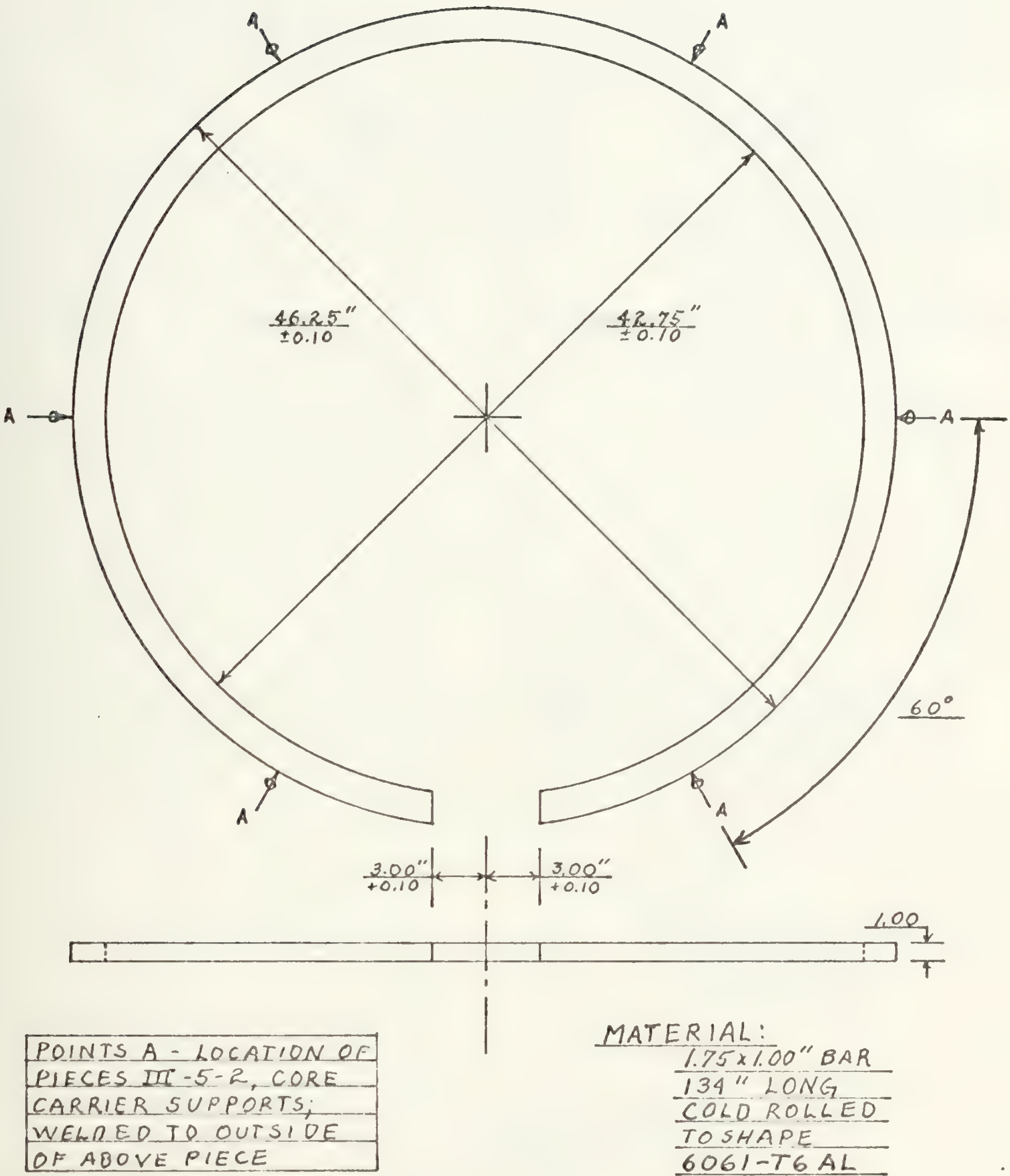


FIGURE III-5-1. PIECE III-5-1, CORE CARRIER

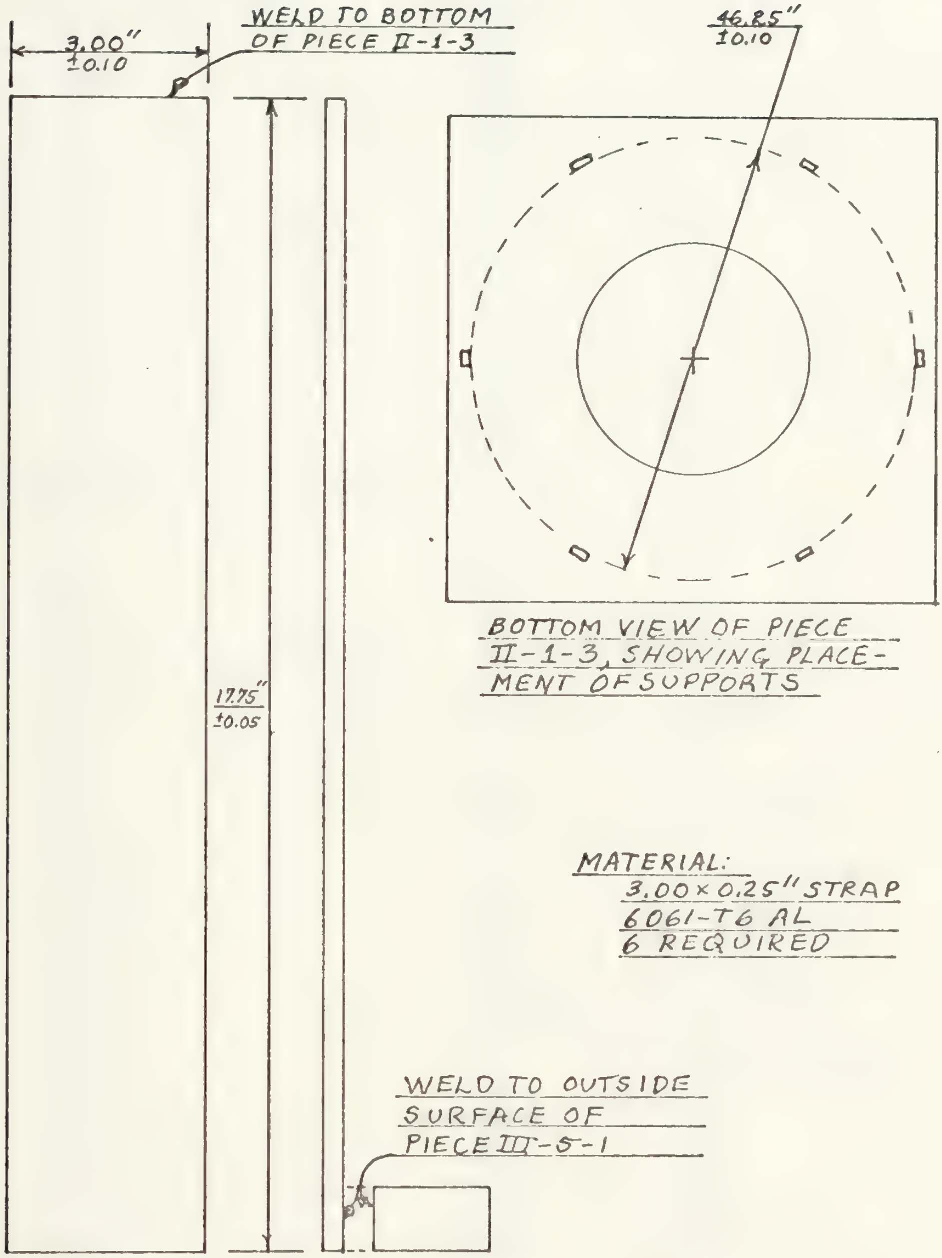


FIGURE III-5-2. PIECE III-5-2, CORE CARRIER SUPPORT

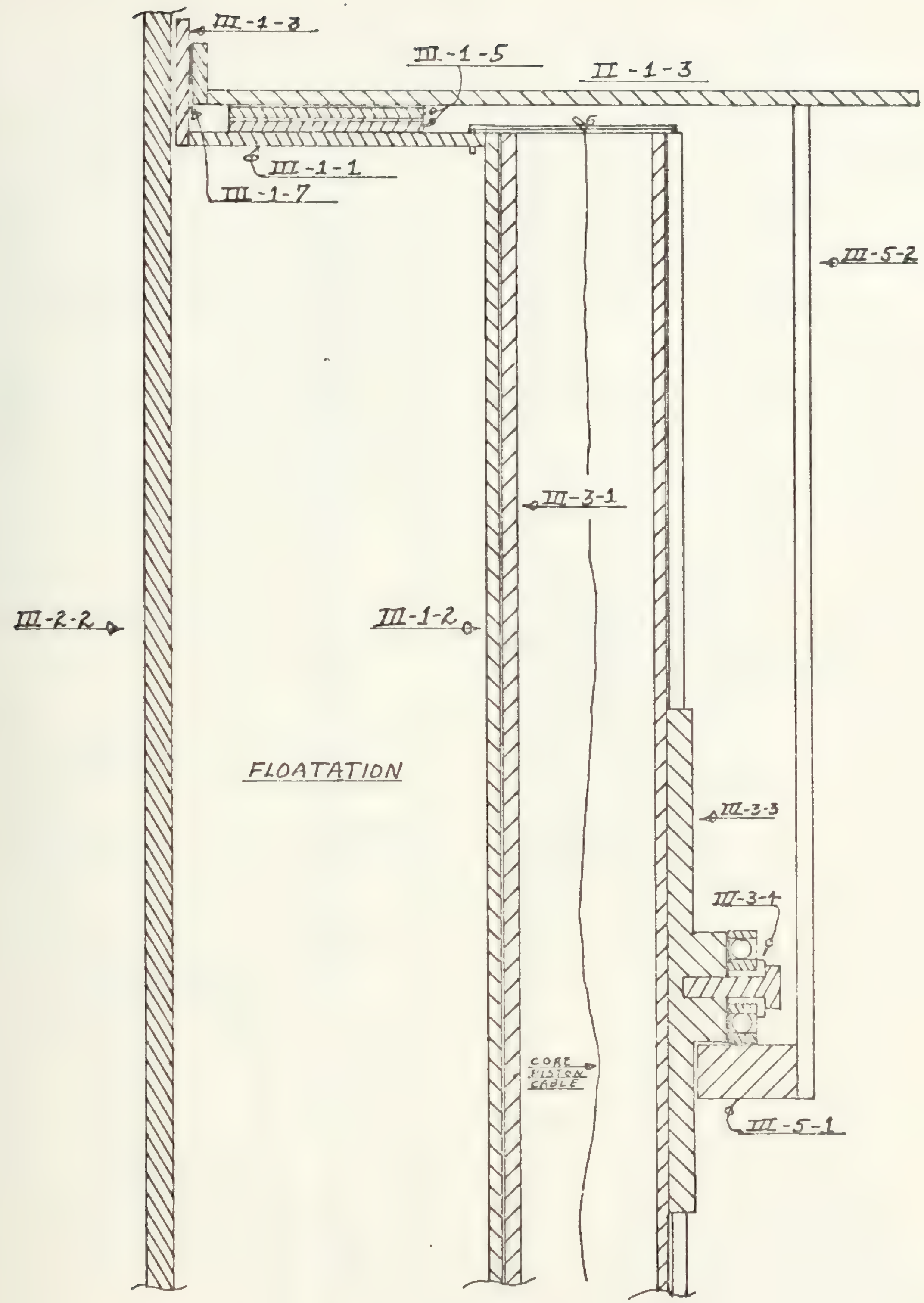


FIGURE III-6-1. CROSS SECTION OF UPPER PORTION OF CORING DEVICE

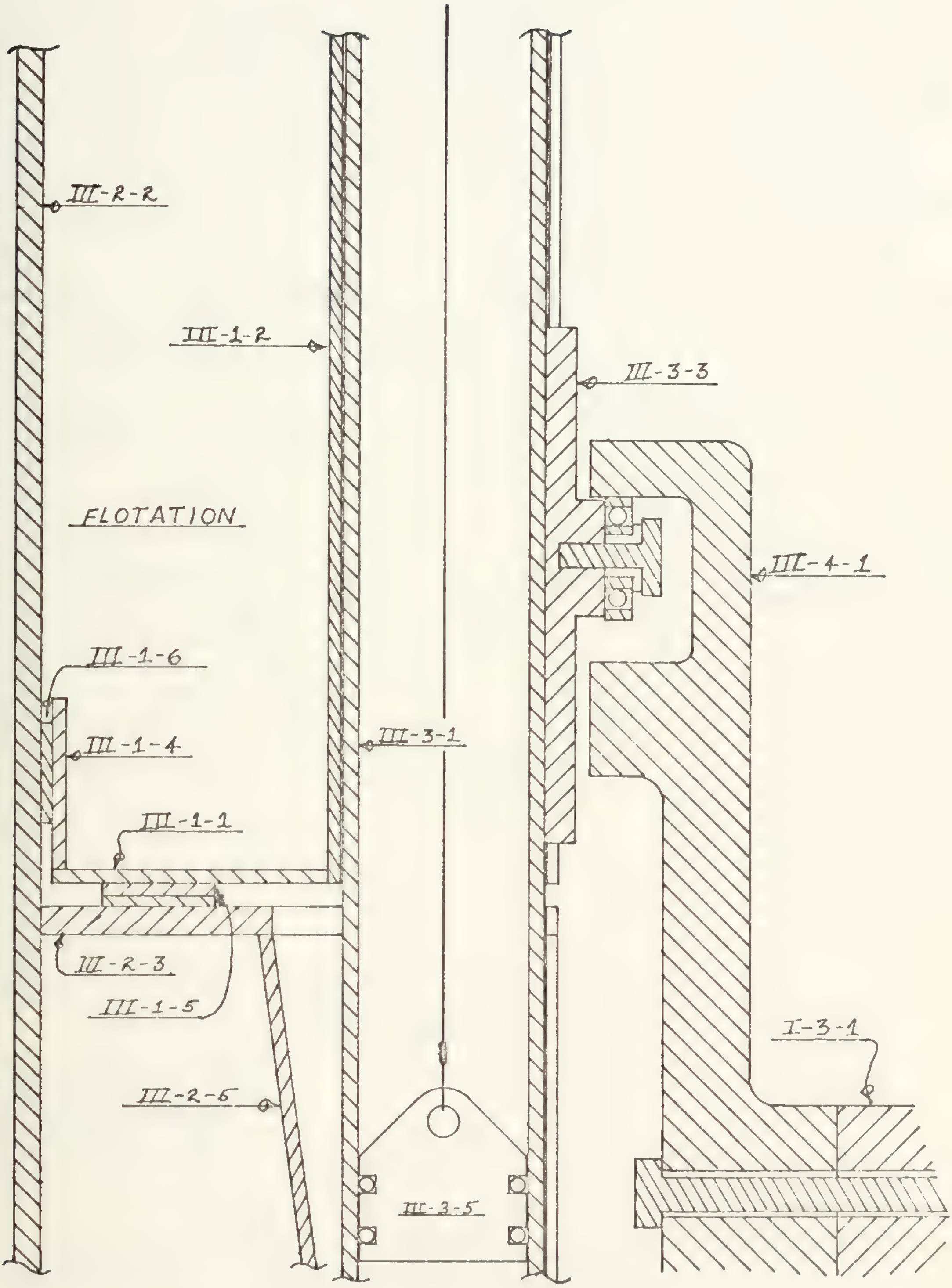


FIGURE III-6-2. CROSS SECTION OF LOWER PORTION OF CORING DEVICE, DURING INSERTION

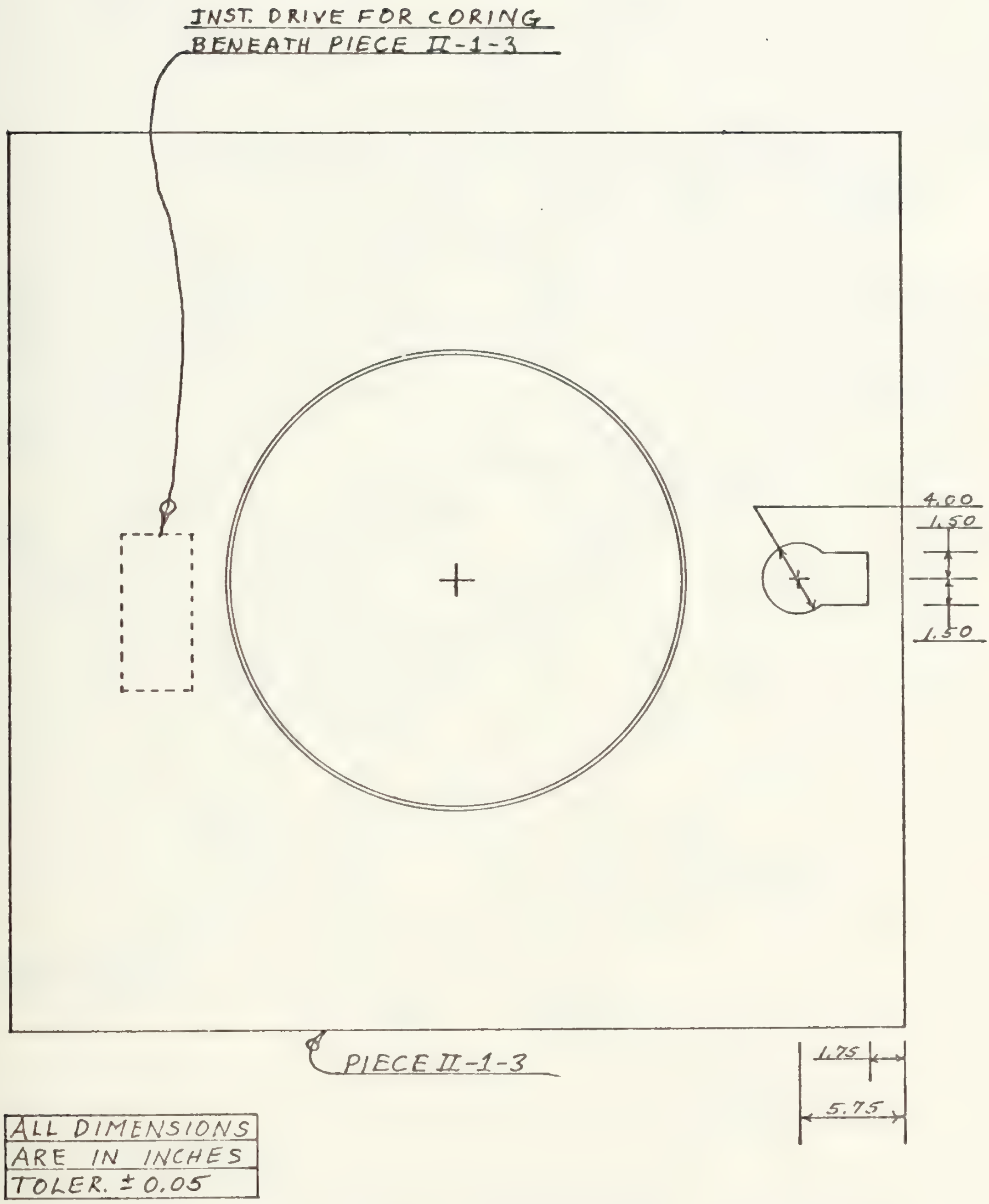


FIGURE III-7. CUT REQUIRED IN PIECE II-1-3 FOR CORE REMOVAL

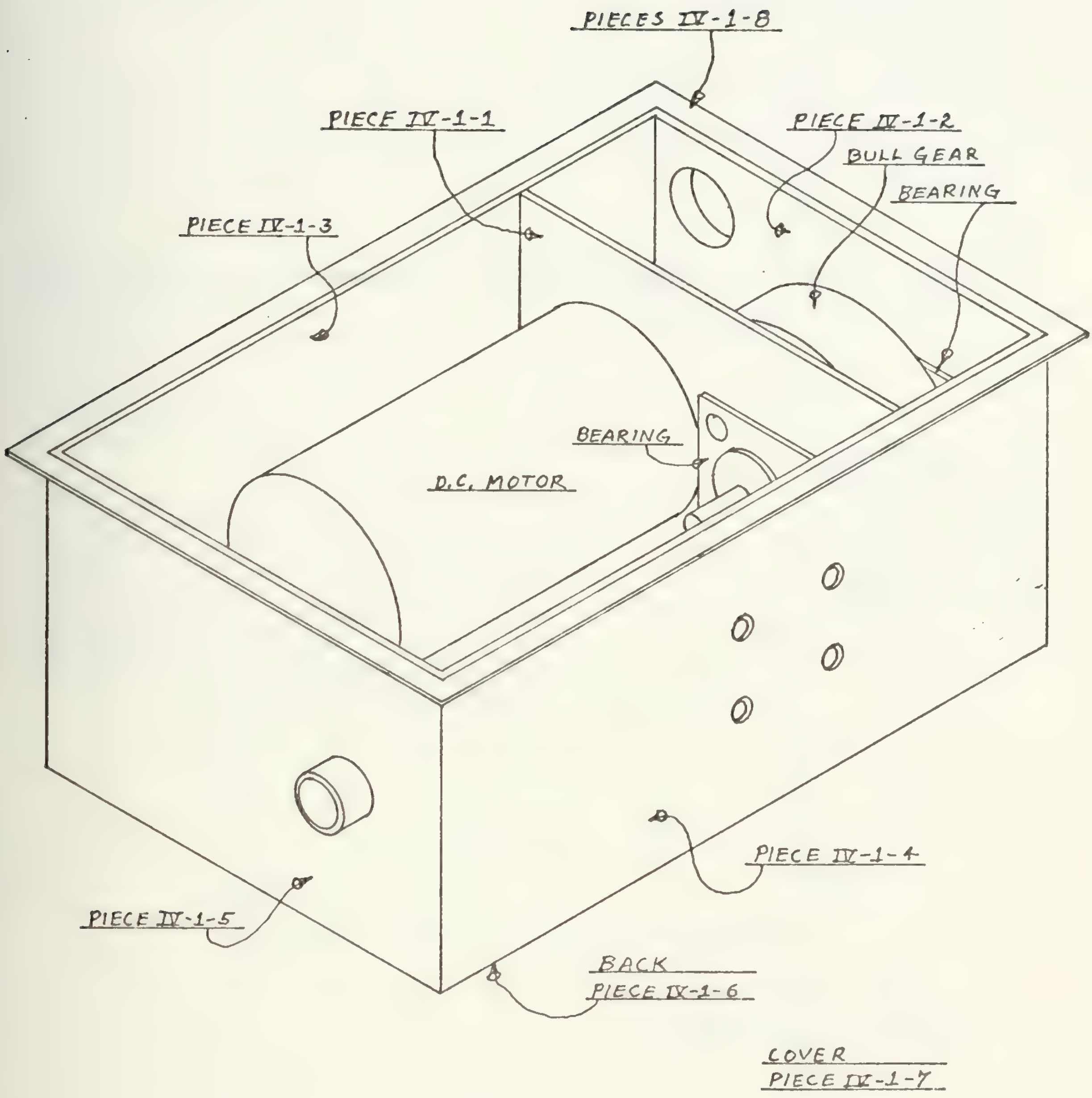


FIGURE IV-1. INSTRUMENT DRIVE MOTOR BOX, WITHOUT COVER

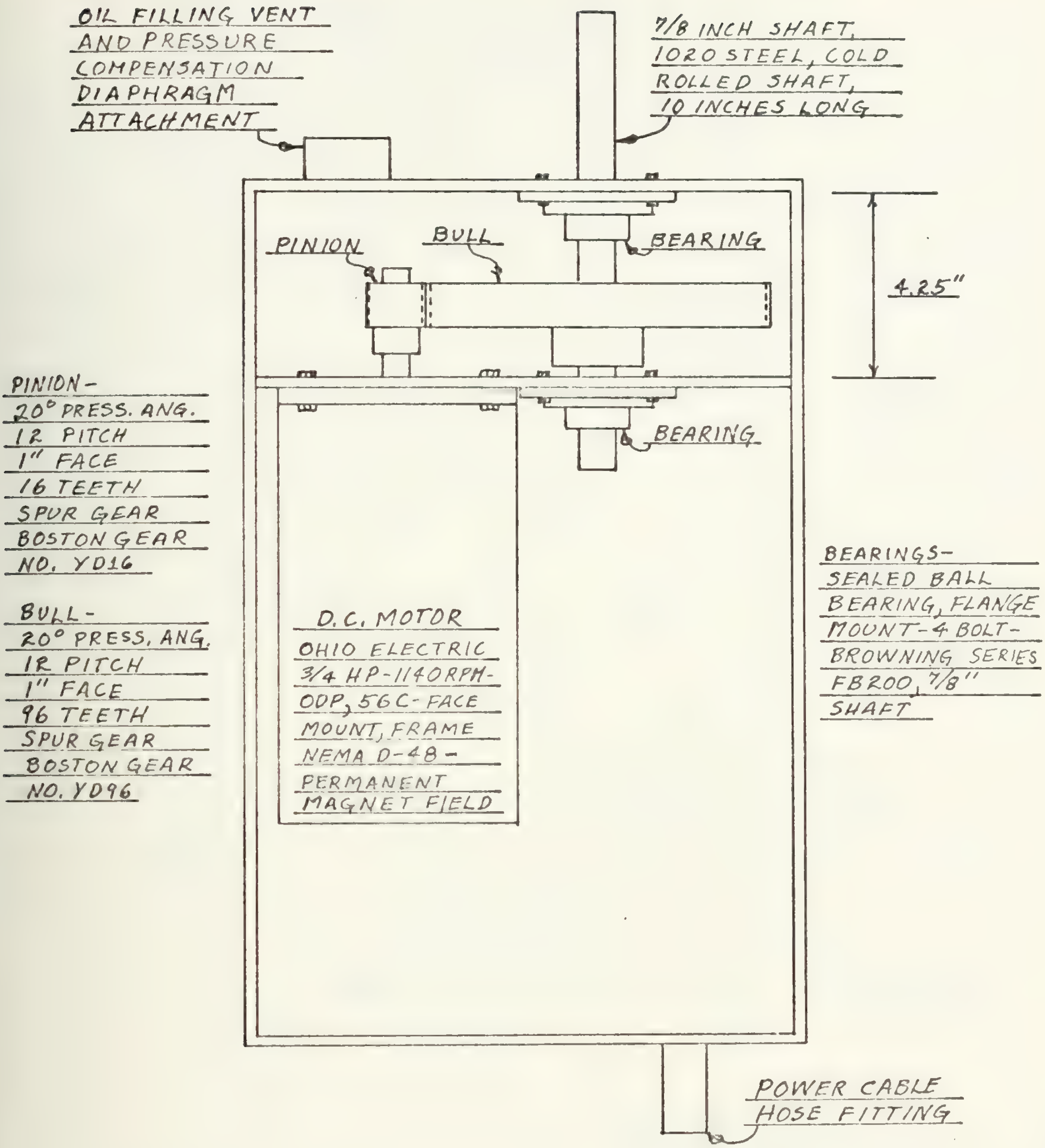
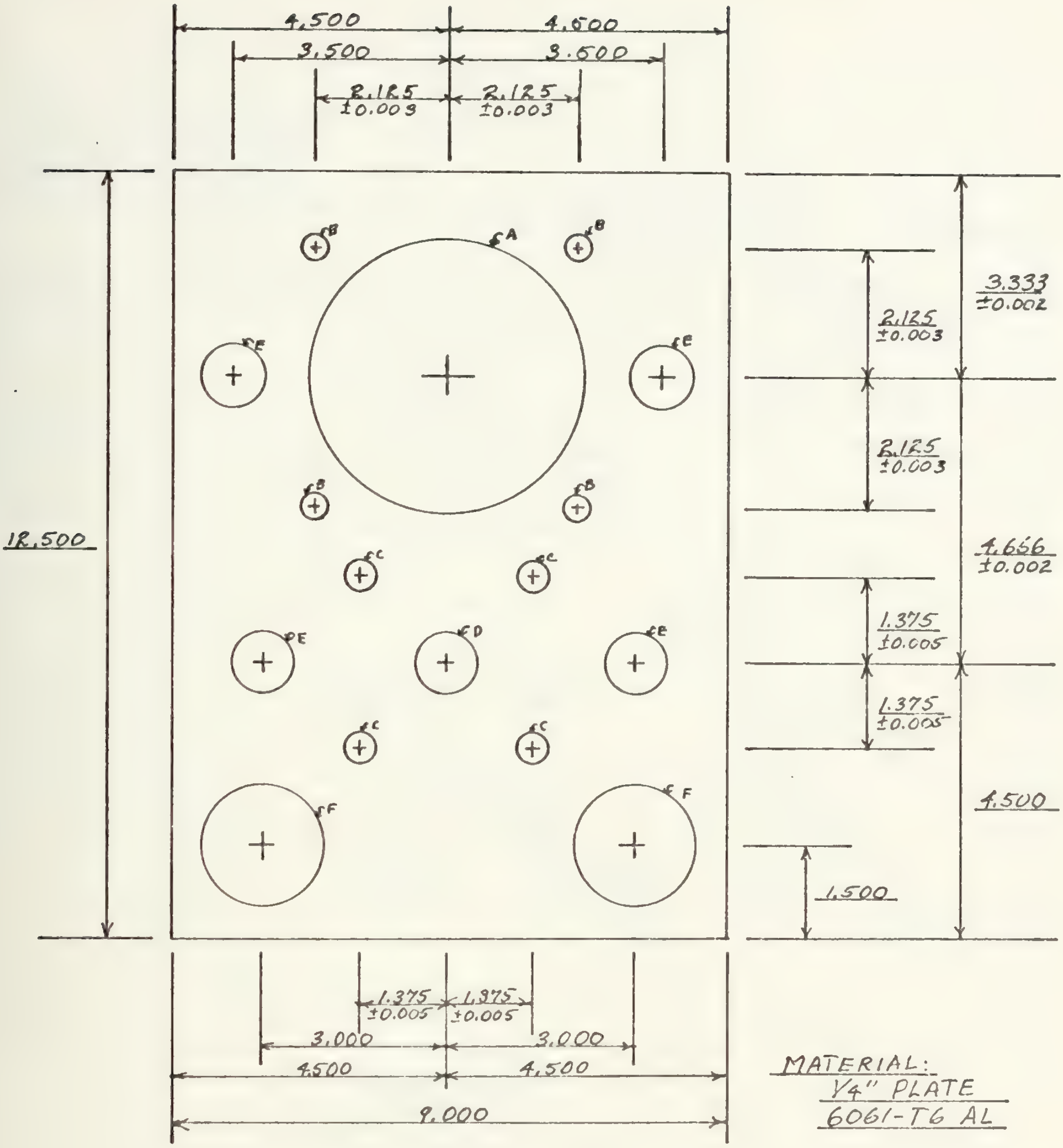


FIGURE IV-1-1. INSTRUMENT DRIVE MOTOR BOX, WITHOUT COVER OR PIECES IV-1-8



ALL DIMENSIONS ARE IN
INCHES, TOLER ±0.010 -
HOLE TOLER ±0.002,
EXCEPT HOLES E & F

HOLE	DIA.	PURPOSE	NO
A	4.500	MOTOR MOUNT	1
B	0.390	BOLT, 3/8-16 UNC, REQ.D	4
C	0.438	BOLT, 7/16 DIA.	4
D	1.000	SHAFT	1
E	1.000	OIL PORT	4
F	2.000	OIL PORT	2

FIGURE IV-1-2. PIECE IV-1-1, MOTOR MOUNT PLATE

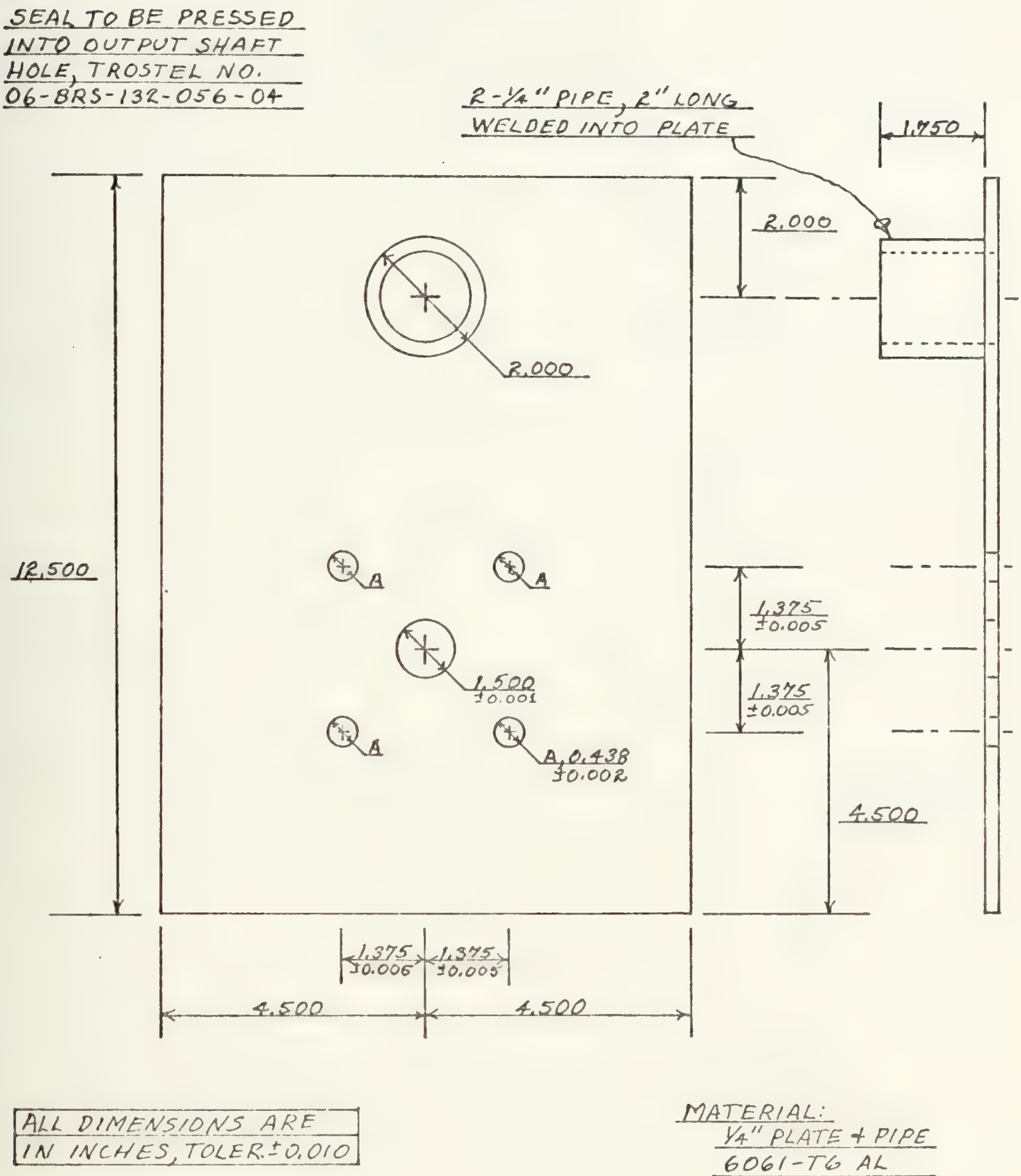


FIGURE IV-1-3. PIECE IV-1-2, TOP PLATE

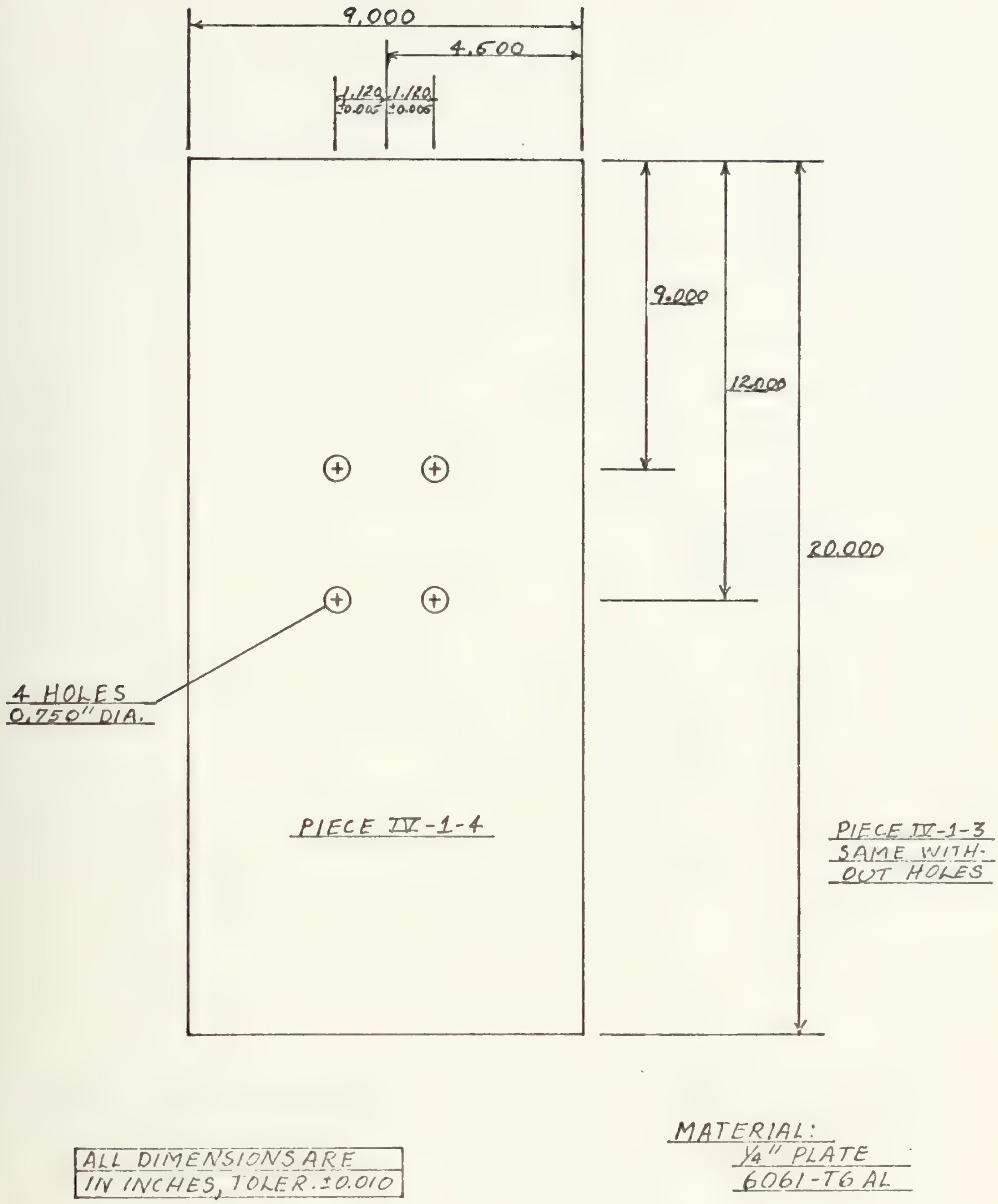


FIGURE IV-1-4. PIECES IV-1-3, 4, SIDES

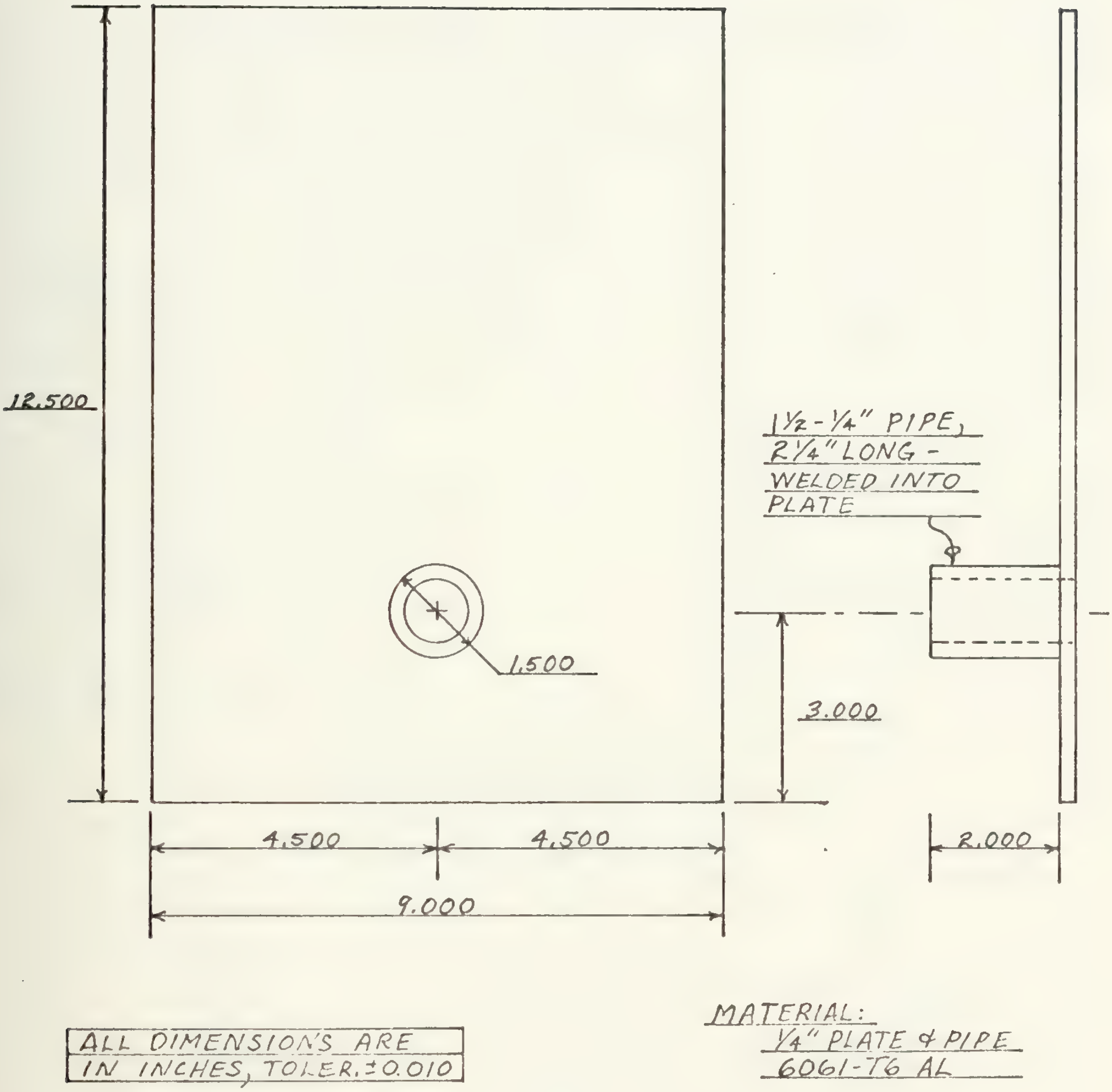


FIGURE IV-1-5. PIECE IV-1-5, BOTTOM

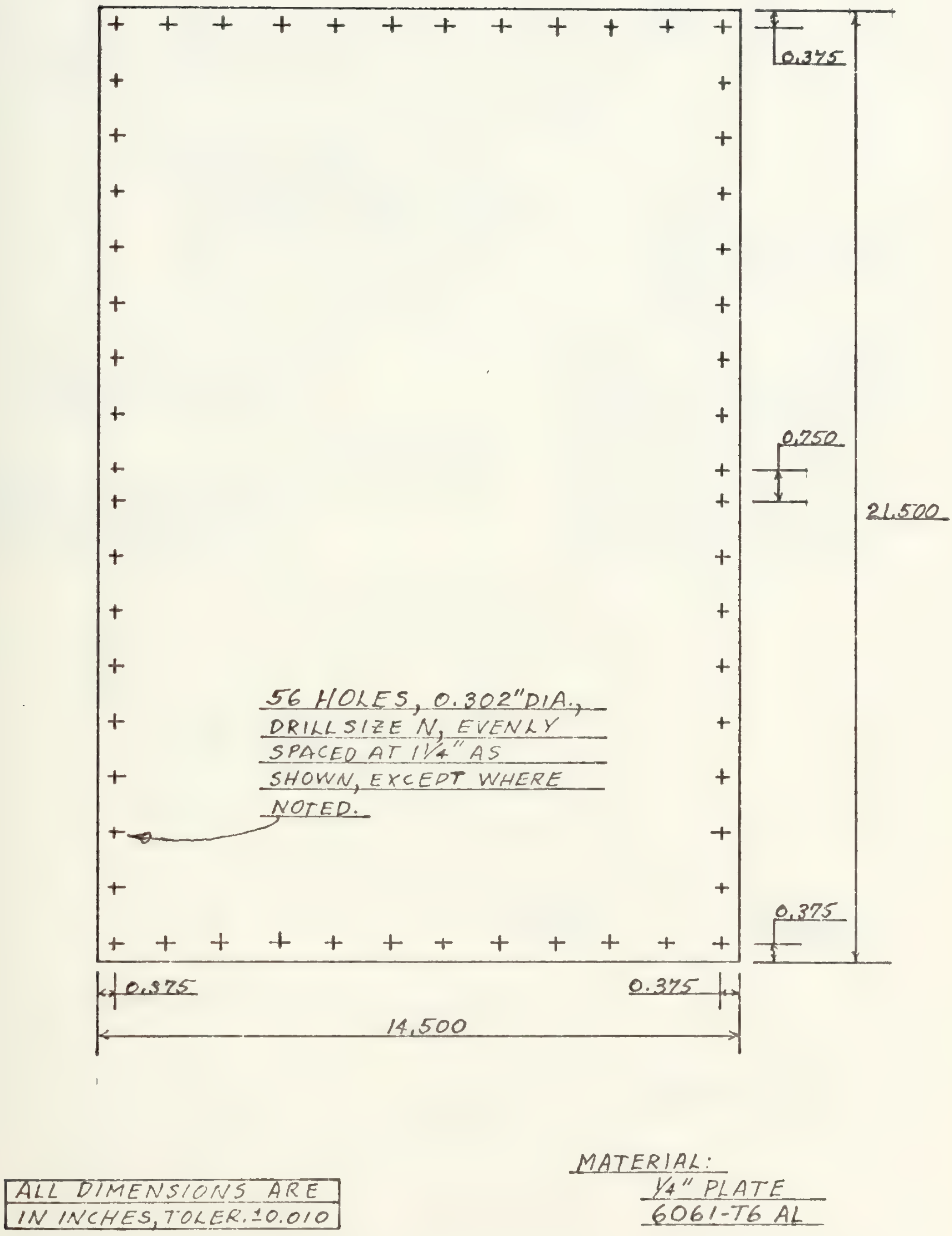
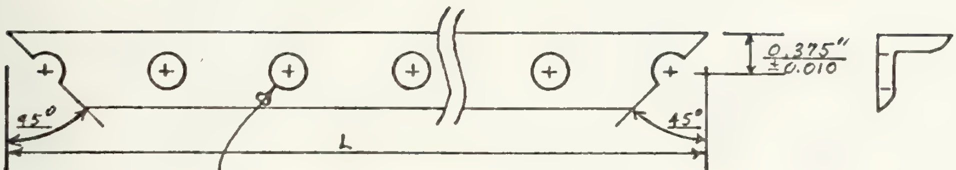


FIGURE IV-1-6. PIECE IV-1-7, COVER



HOLES, BORED AND
TAPPED FOR 1/4-20 UNC
THREADS, PATTERN OF
FIG IV-1-7

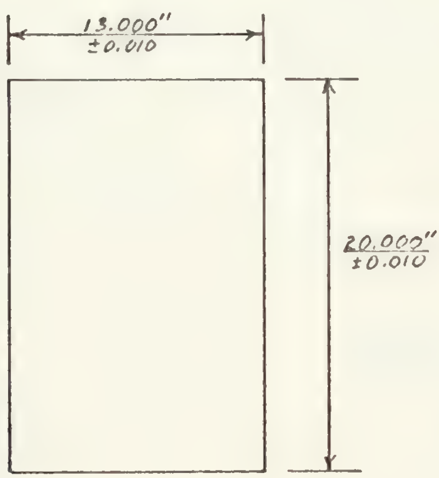
MATERIAL:
3/4"-EQUAL ANGLE
A.A. STD.
6061-T6 AL

2 EACH, L = 14.500"
2 EACH, L = 21.500"

PIECES IV-1-8

PIECES TO BE WELDED
TO OUTSIDE OF BOX-
SEE FIG IV-1

REQUIRES 56 1/4-20 UNC
BOLTS TO SECURE COVER,
RECOMMEND MOLDED
NYLON BOLTS WITH HEX
HEADS, GRIES NO. 1/4-20
UNC-HEX-NYLON, USE
A SILICONE SEAL BET-
WEEN PIECES IV-1-8+7.



MATERIAL:
1/4" PLATE
6061-T6

PIECE IV-1-6

FIGURE IV-1-7. PIECES IV-1-6,8,
BACK AND RETAINER

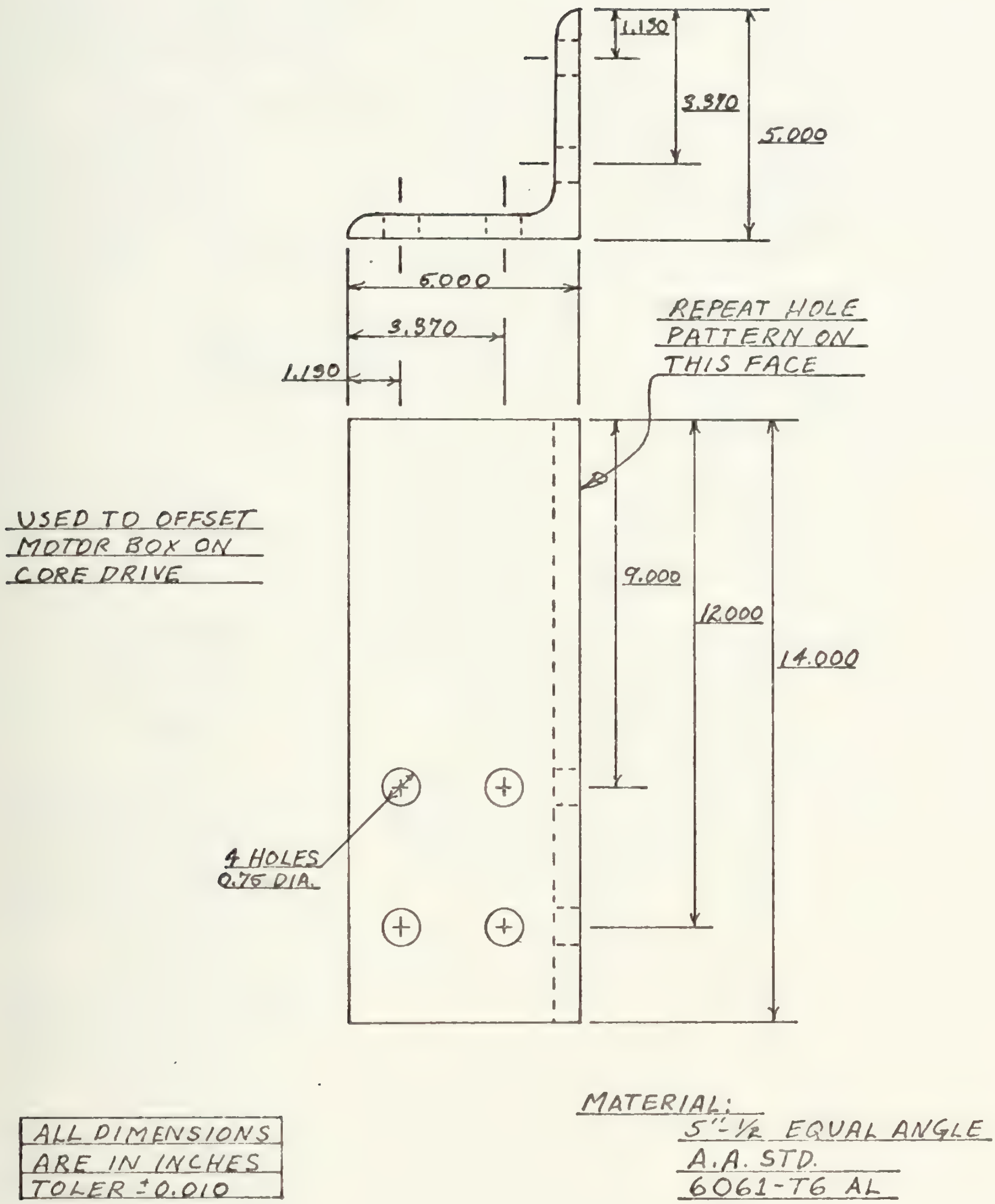


FIGURE IV-1-8. PIECE IV-1-9, MOTOR BOX OFFSET

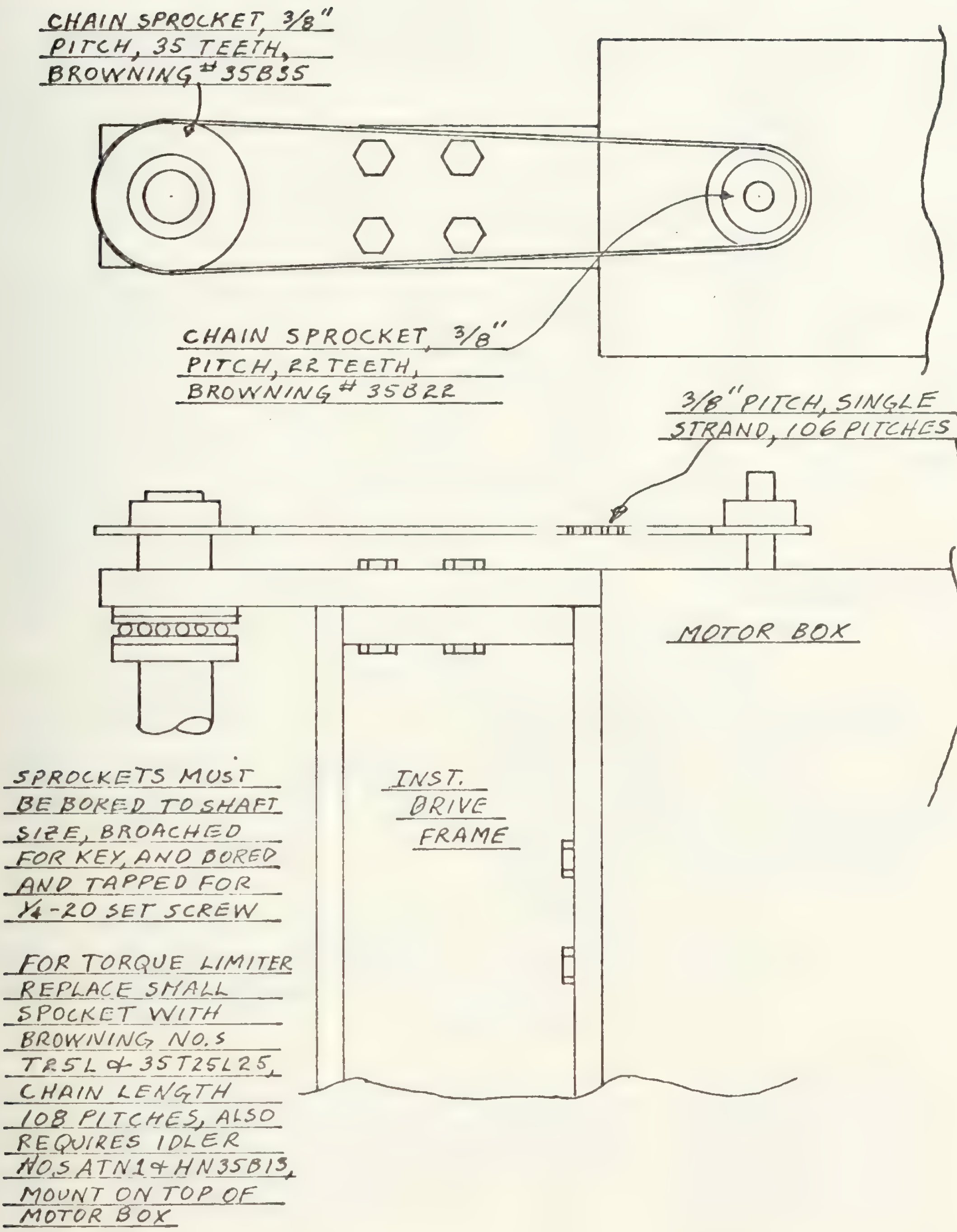


FIGURE IV-1-9. CHAIN DRIVE

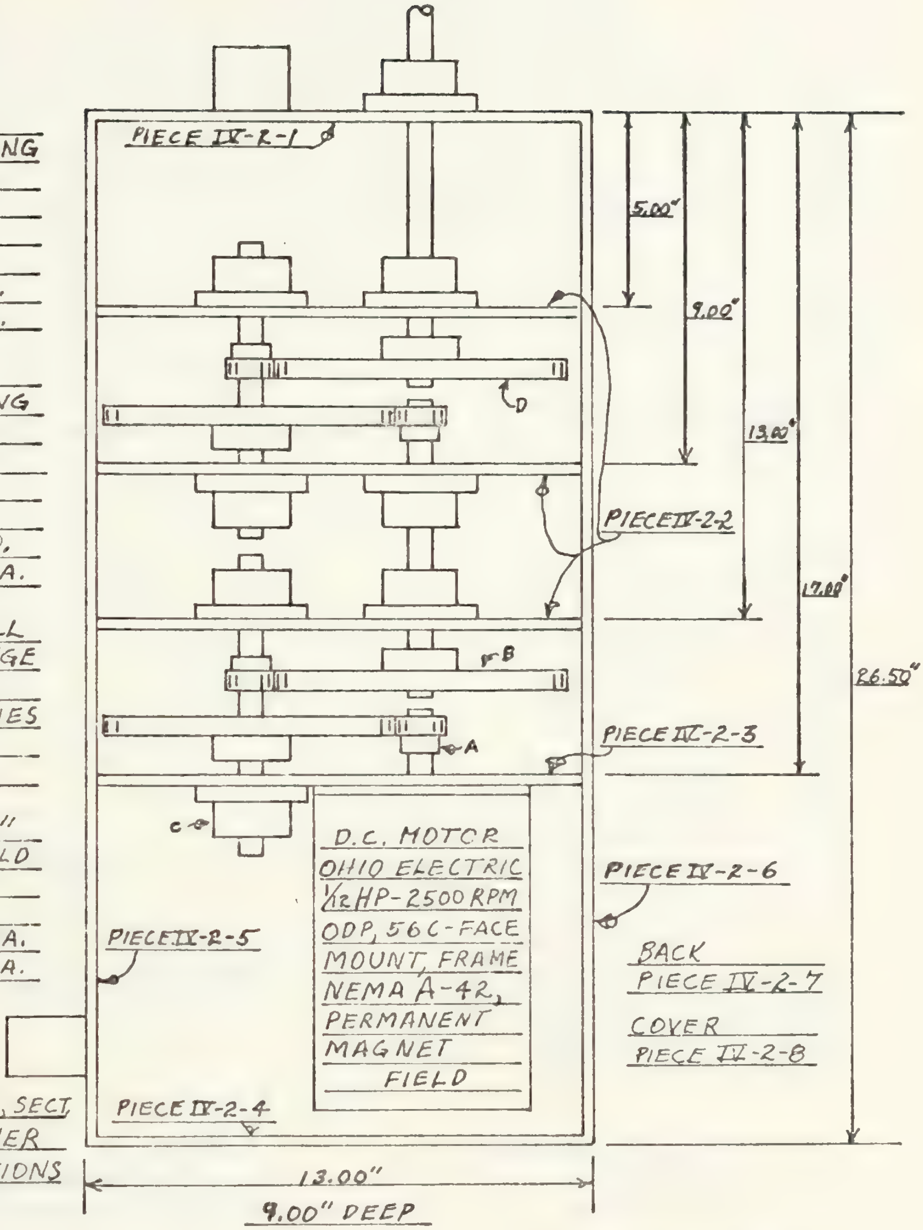
A - GEAR
14 1/2° PRESS. ANG
16 PITCH
1/2" FACE
20 TEETH
SPUR GEAR
BROWNING NO.
NSS1620, 4 EA.

B - GEAR
14 1/2° PRESS. ANG
16 PITCH
1/2" FACE
120 TEETH
SPUR GEAR
BROWNING NO.
NCS16120, 4 EA.

C - SEALED BALL
BEARING - FLANGE
MOUNT, 4 BOLT,
BROWNING SERIES
FB200, 5/8"
SHAFT, 8 EA.

SHAFTING - 5/8"
1020 STEEL, COLD
ROLLED AND
POLISHED
7.50" LONG - 3 EA.
16.00" LONG - 1 EA.

SEE APPENDIX A, SECT.
D-2 FOR OTHER
RECOMMENDATIONS



GEARS MUST BE REAMED TO A SMOOTH FIT ON THE SHAFT, BROACHED FOR A 3/16" SQ. KEY, AND DRILLED AND TAPPED FOR A 1/4-20UNC SET SCREW. SHAFTS MUST BE MILLED FOR A 3/16" SQ. KEY IN THE AREA OF THE GEARS

D - TORQUE LIMITER,
BROWNING NO. T35L,
GEAR, SPECS. OF "B",
BROWNING NO. NCG-16120, BORED AND
MACHINED TO FIT
TORQUE LIMITER

FIGURE IV-2. CORE CYLINDER MOTOR BOX

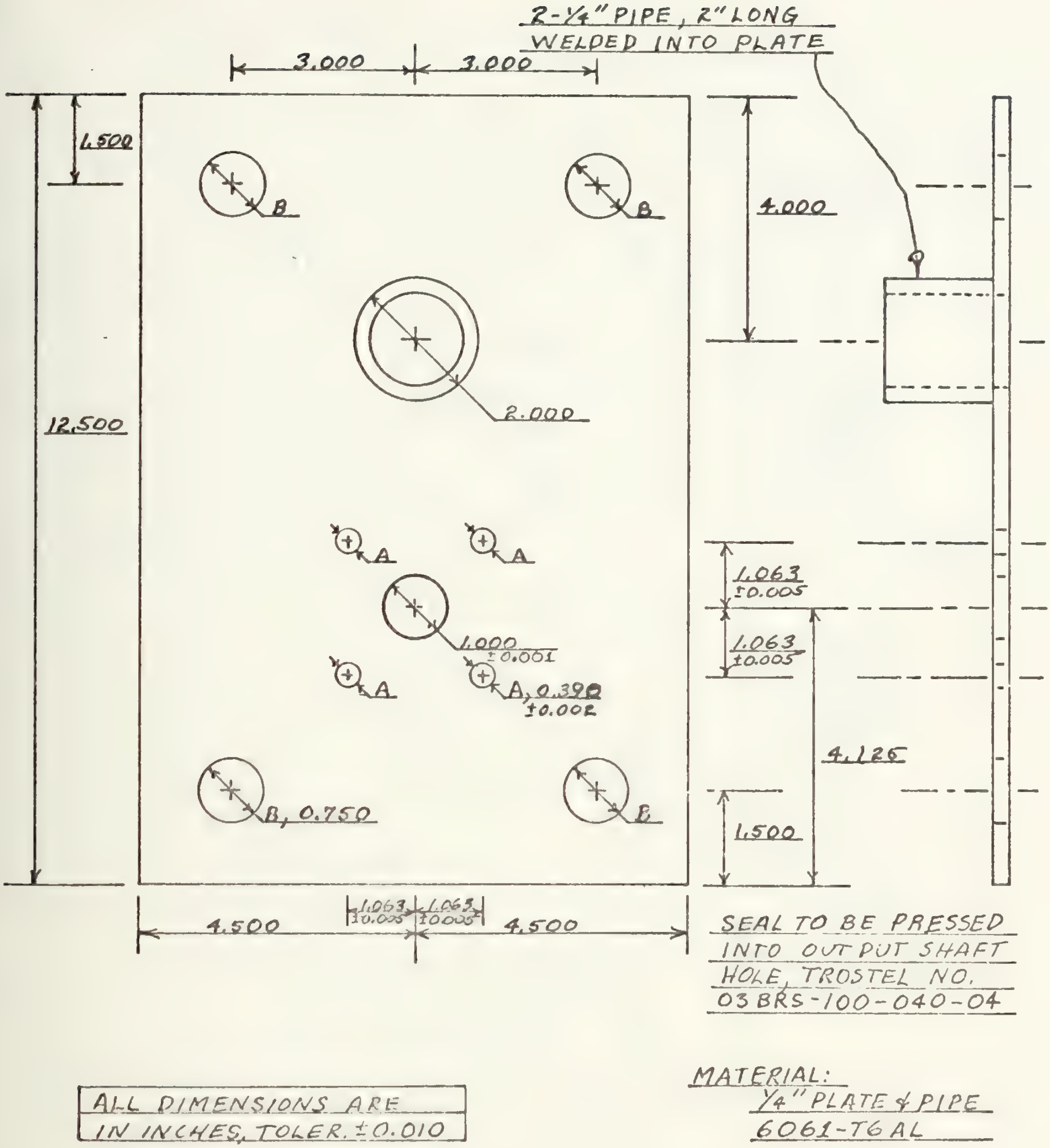


FIGURE IV-2-1. PIECE IV-2-1, TOP PLATE

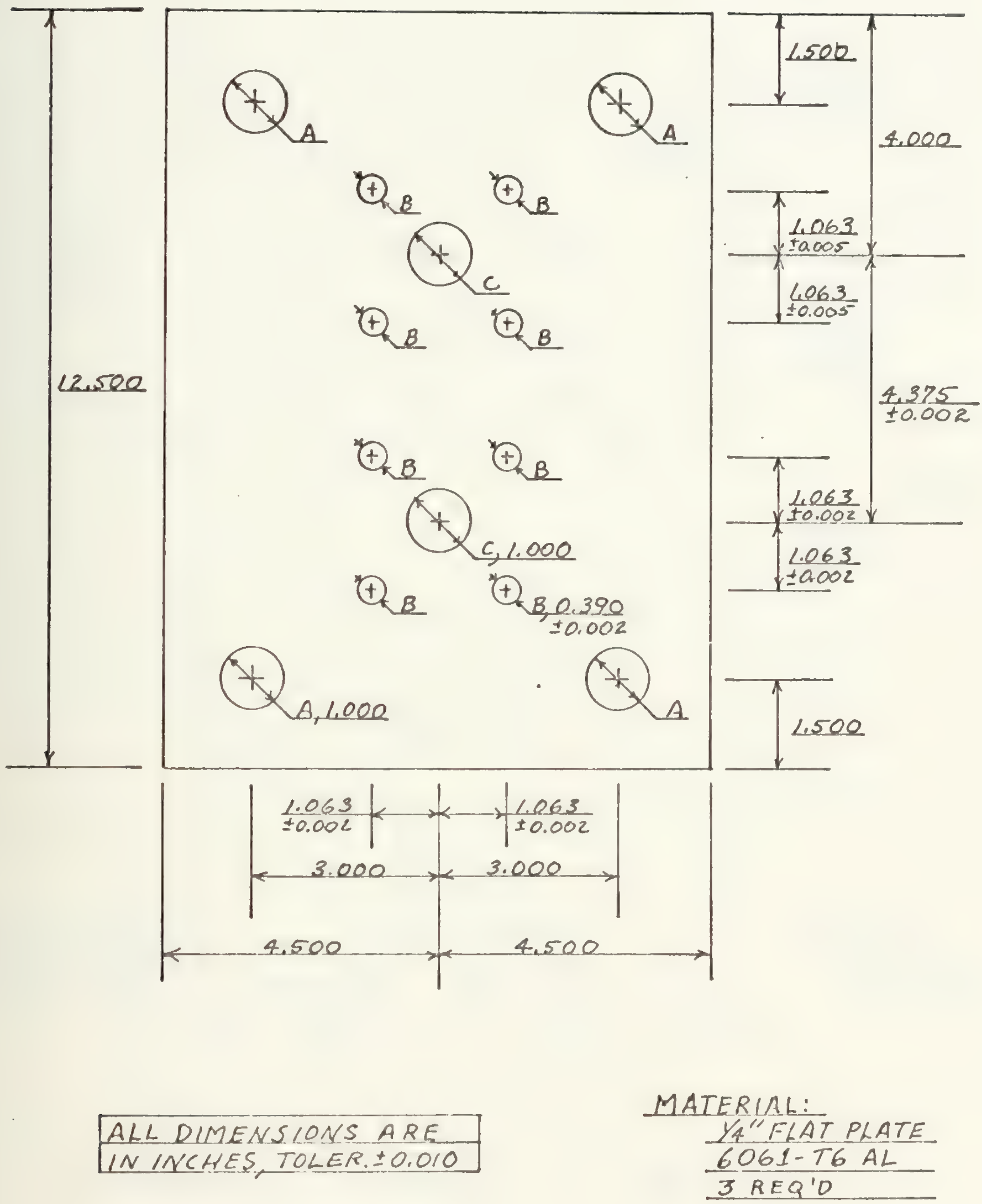


FIGURE IV-2-2. PIECE IV-2-2, CHANGE GEAR PLATE

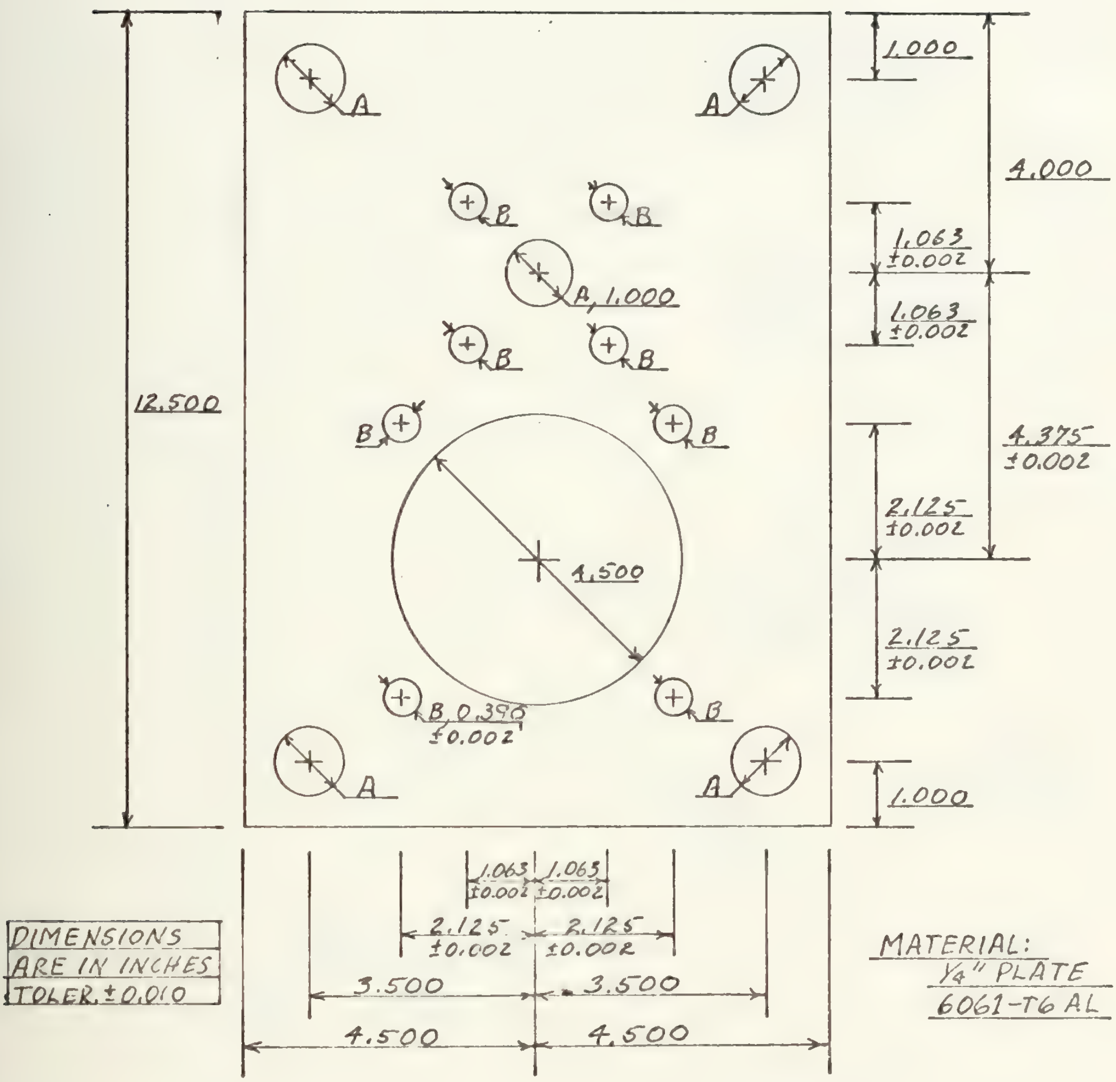


FIGURE IV-2-3. PIECE IV-2-3, MOTOR MOUNT PLATE

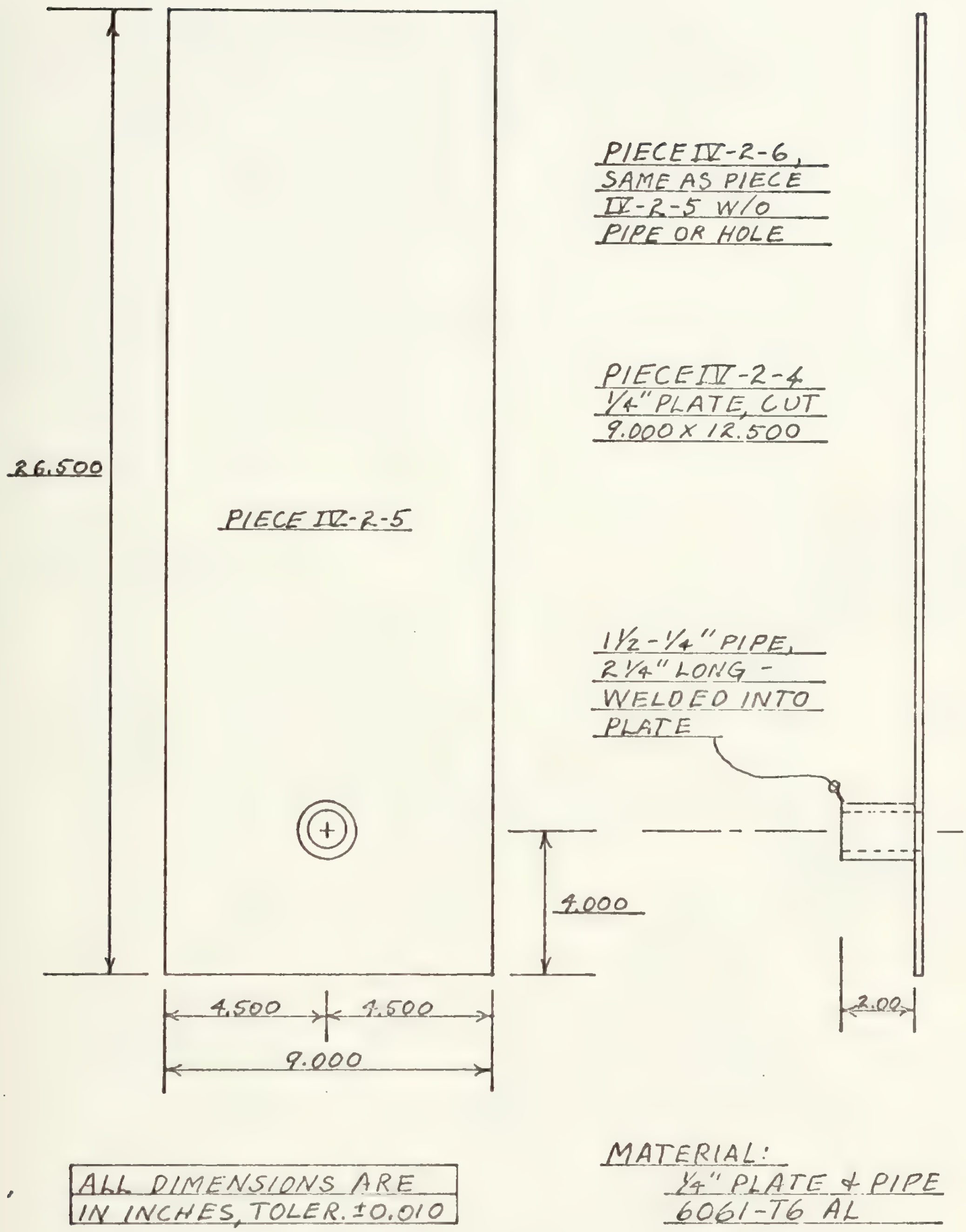


FIGURE IV-2-4. PIECES IV-2-4,5,6,
SIDE AND BOTTOM PLATES

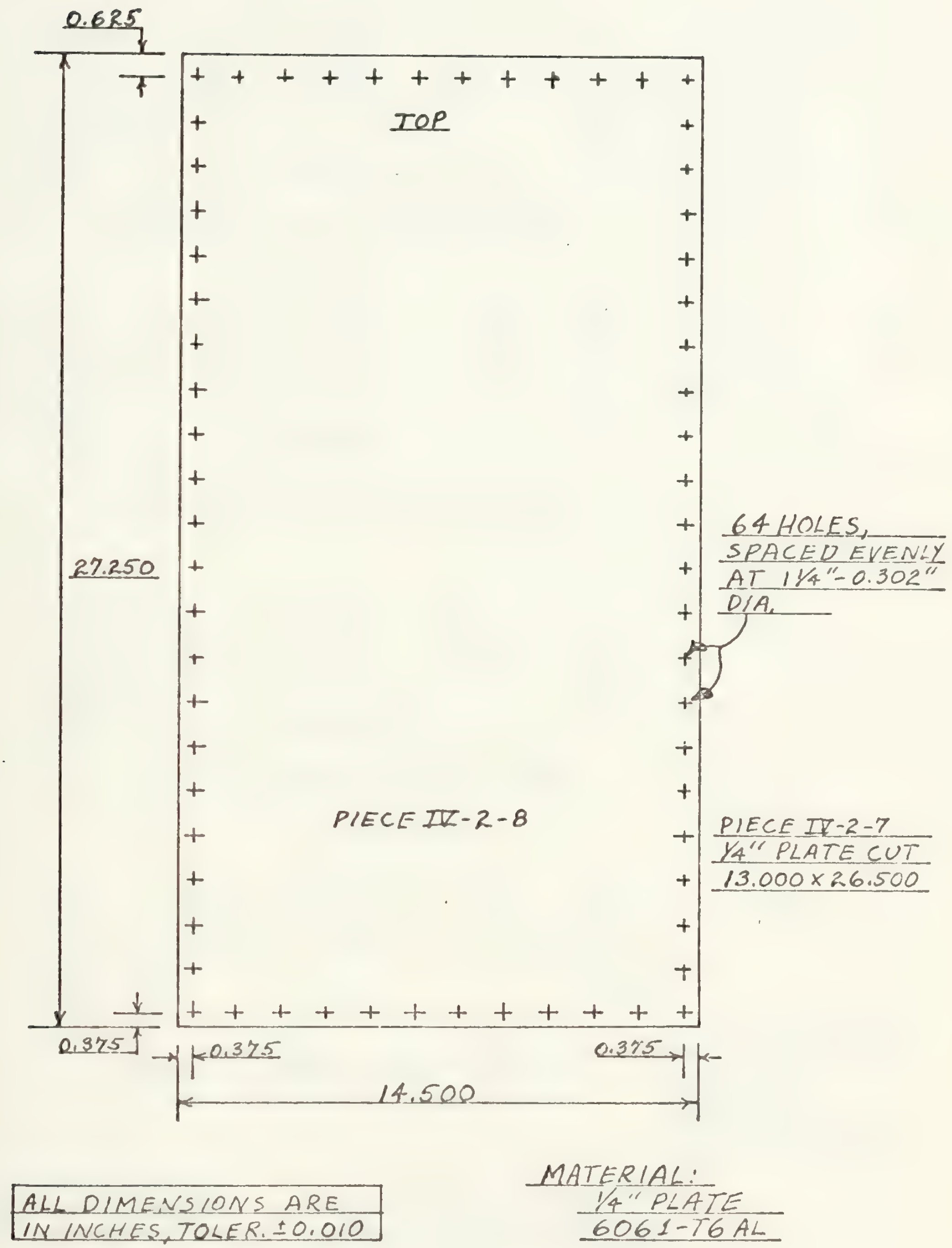


FIGURE IV-2-5. PIECES IV-2-7,8,
BACK AND COVER PLATES

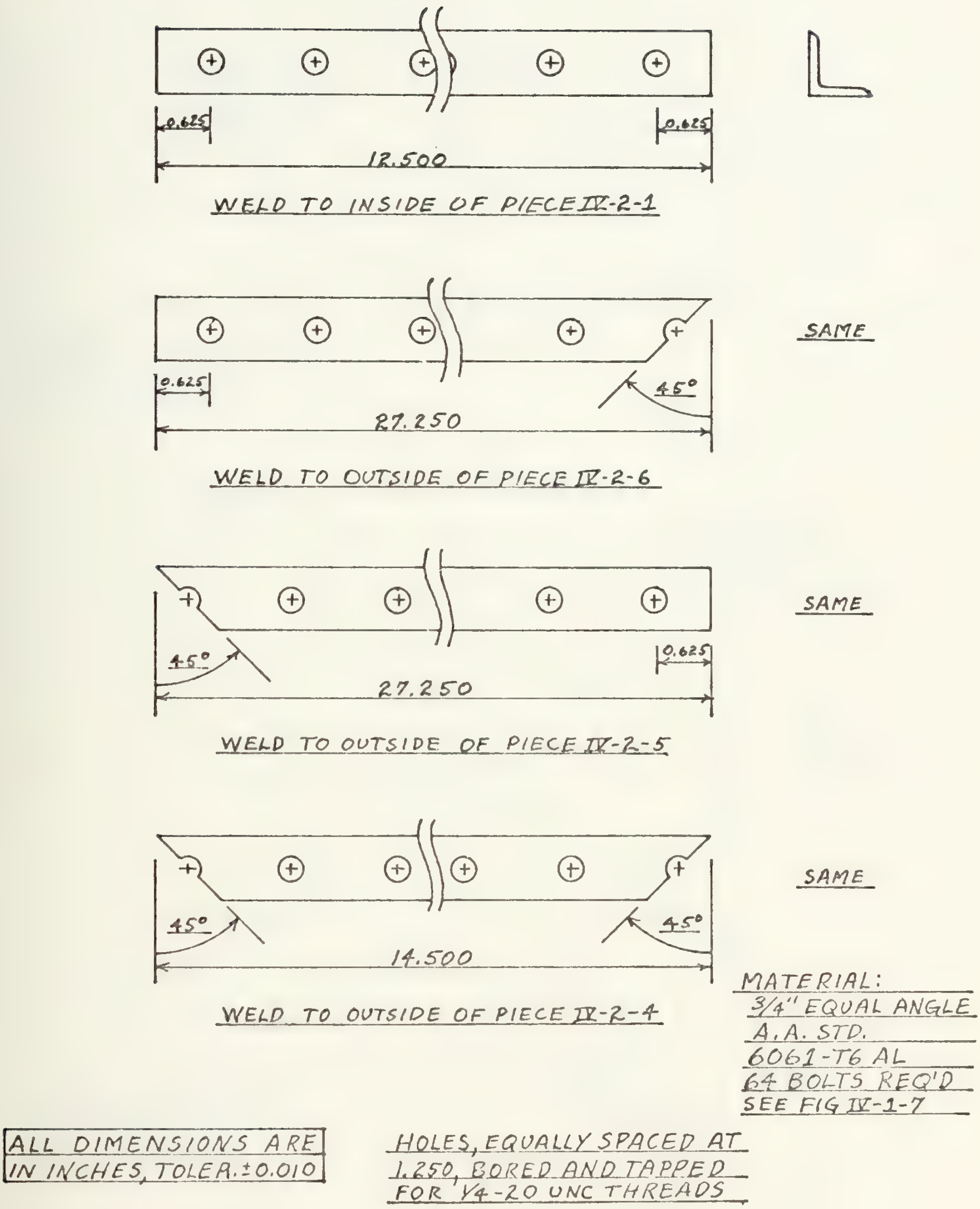


FIGURE IV-2-6. PIECES IV-2-9, RETAINERS

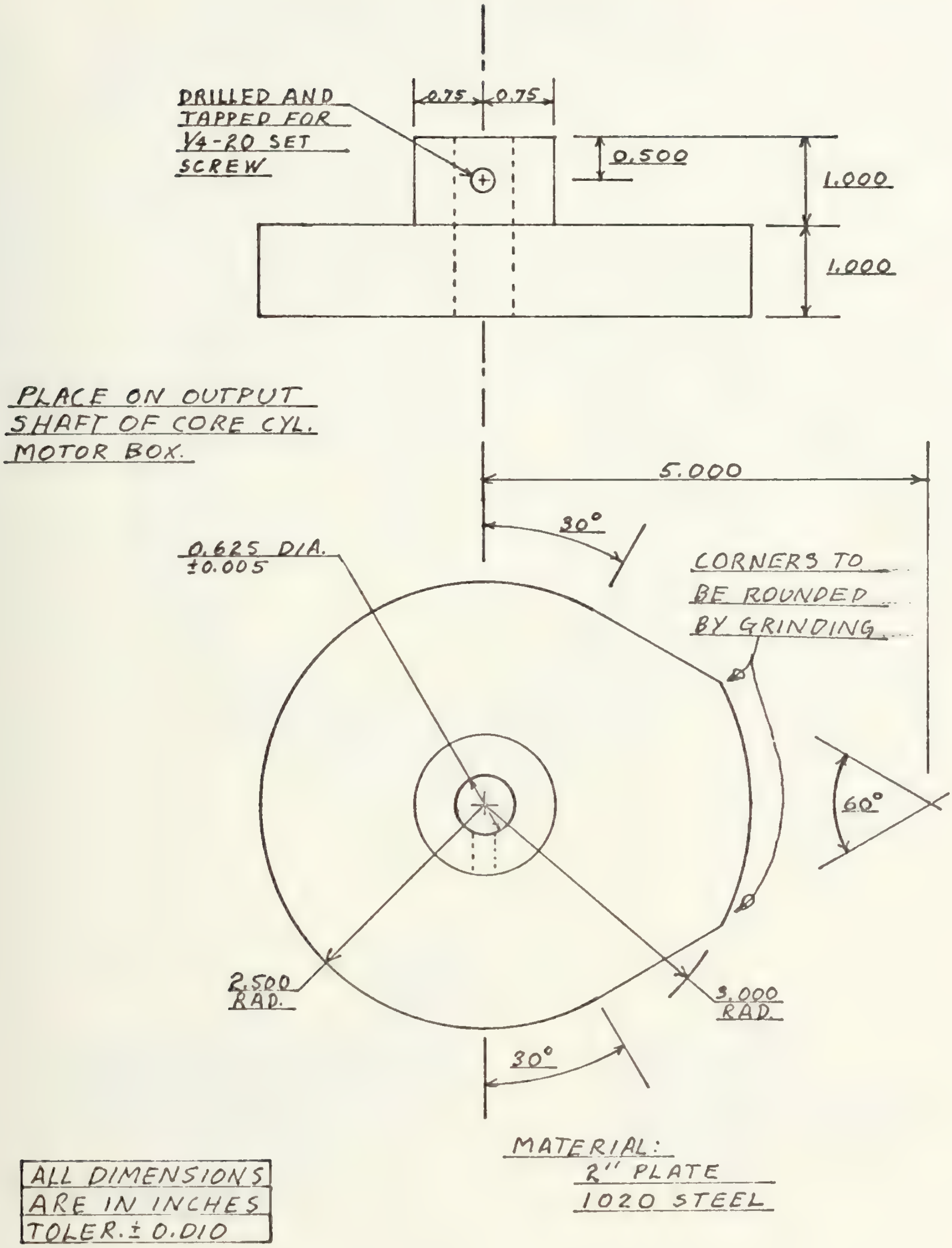
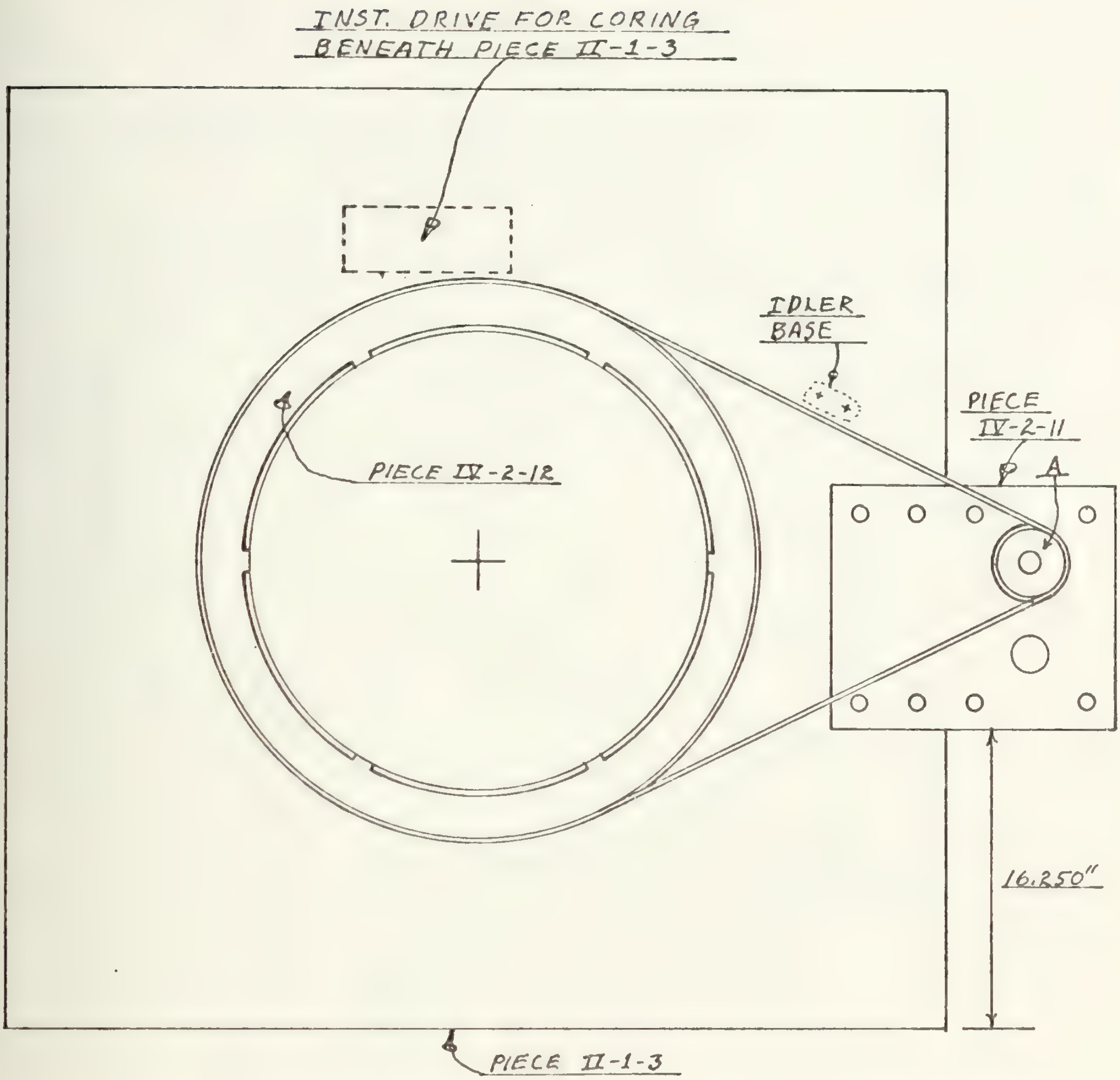


FIGURE IV-2-7. PIECE IV-2-10, SHUT OFF CAM



A-CHAIN SPROCKET, 1" PITCH,
12 TEETH, BROWNING # 80P12,
TAPER BUSHING WITH 1/8 x 1/16
KEYWAY, BORE 5/8"

PIECE II-1-3 TO BE BORED
TO ACCEPT PIECE IV-2-11

1" PITCH, SINGLE STRAND,
120 PITCHES

IDLER REQUIRED, BROWNING
NO.S ATN2 & HN80B13

FIGURE IV-2-8. CHAIN DRIVE

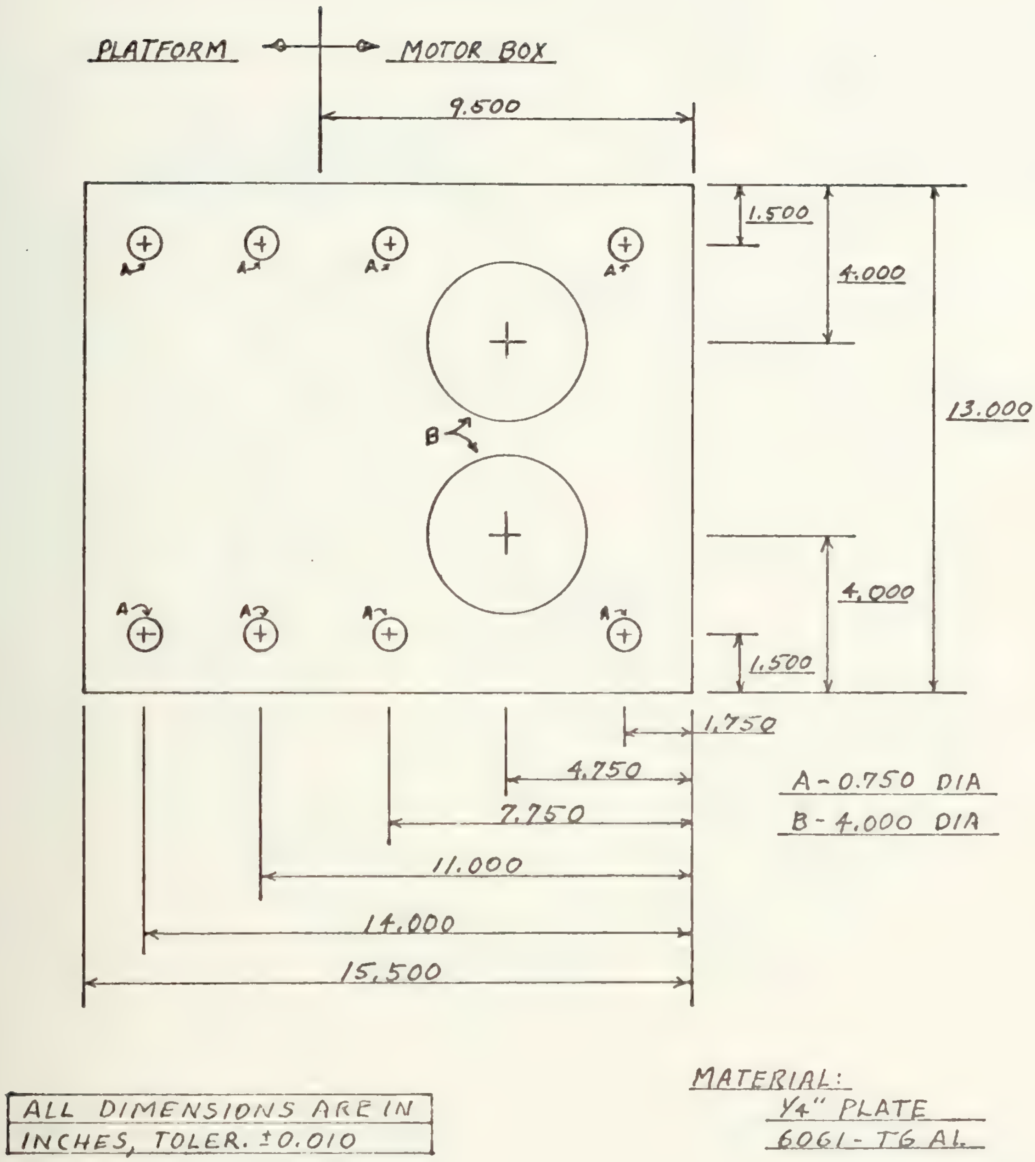


FIGURE IV-2-9. PIECE IV-2-11, CONNECTING PLATE

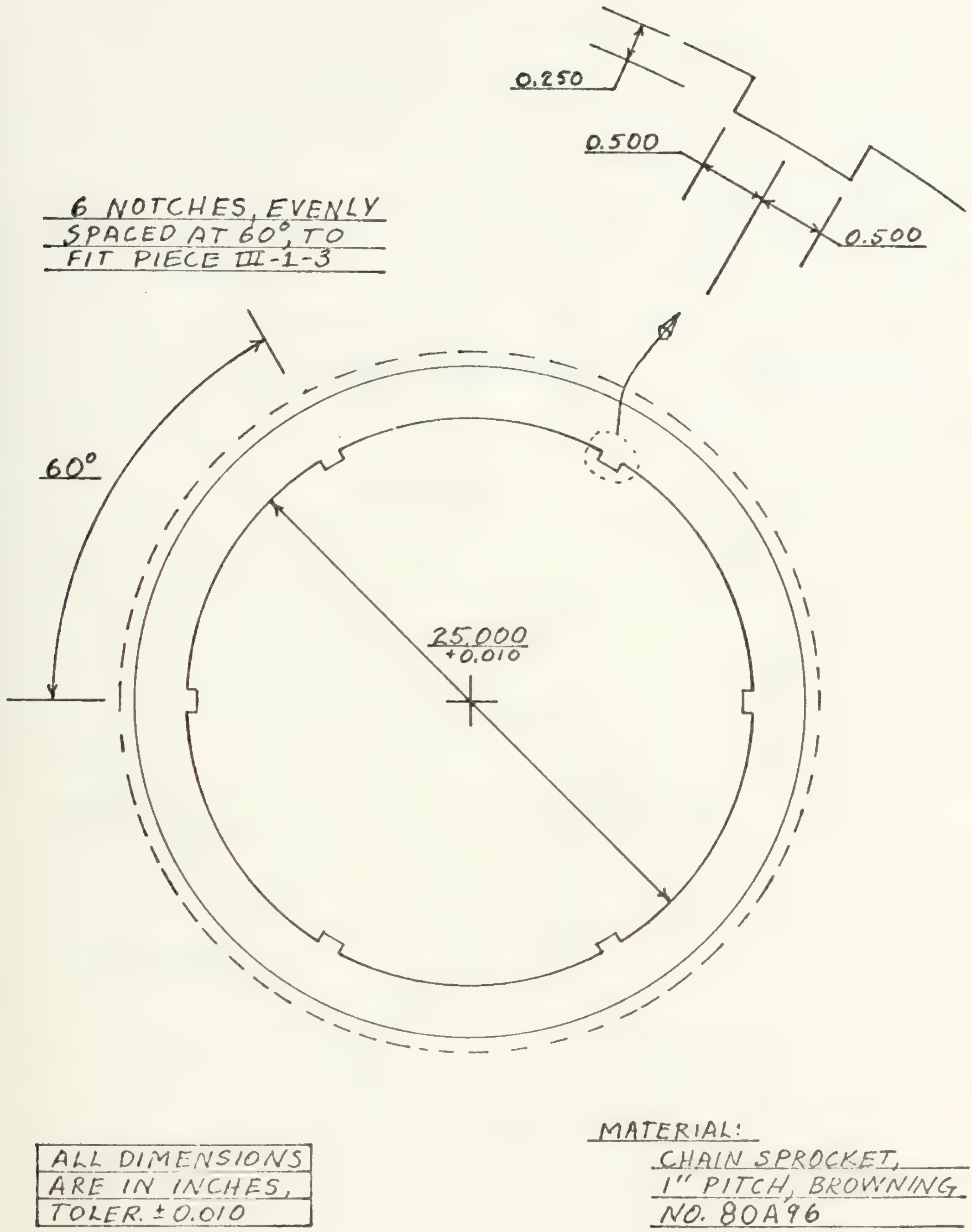


FIGURE IV-2-10. PIECE IV-2-12, CORING CYLINDER SPROCKET

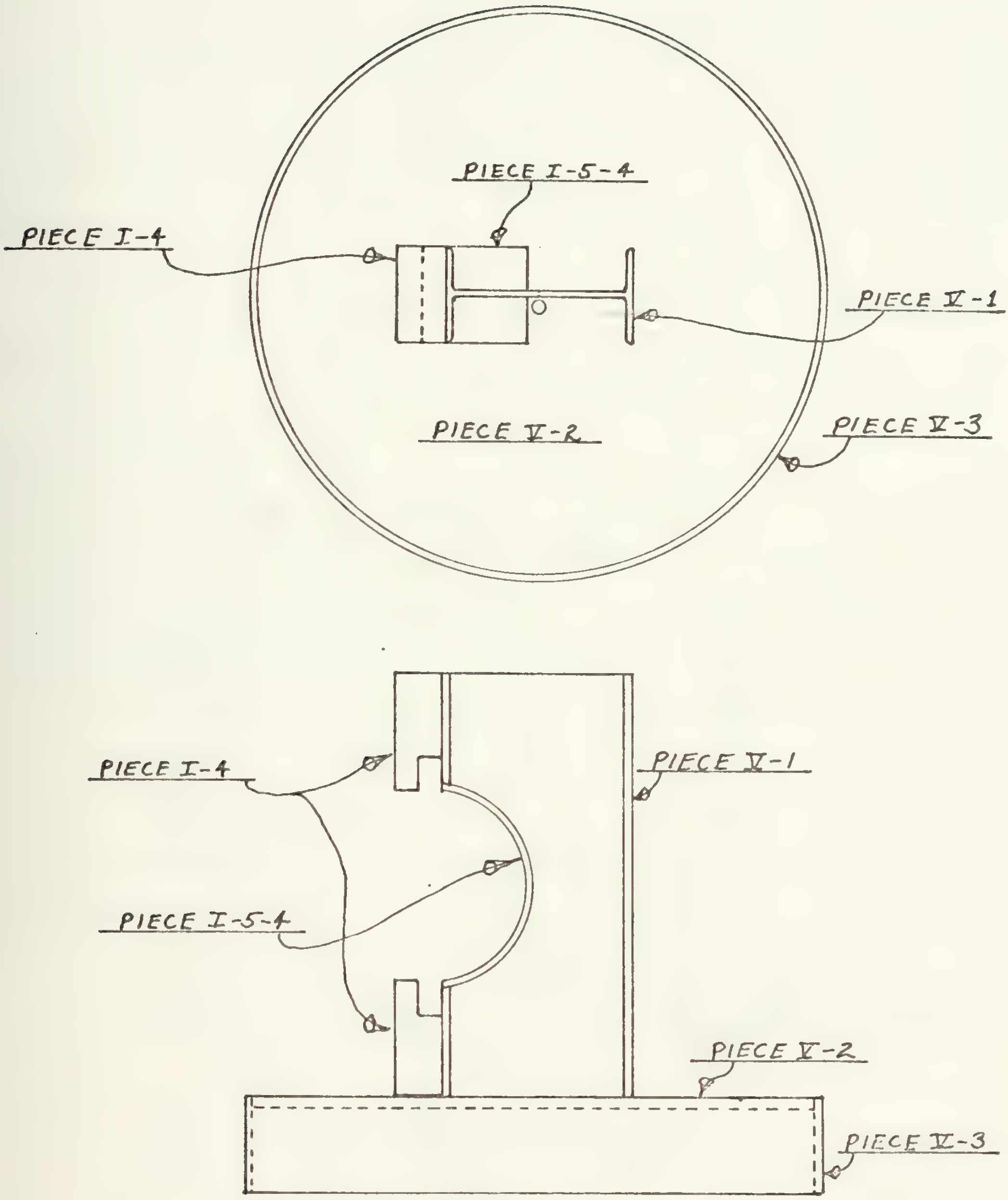


FIGURE V-1. HOLD DOWN PLATE

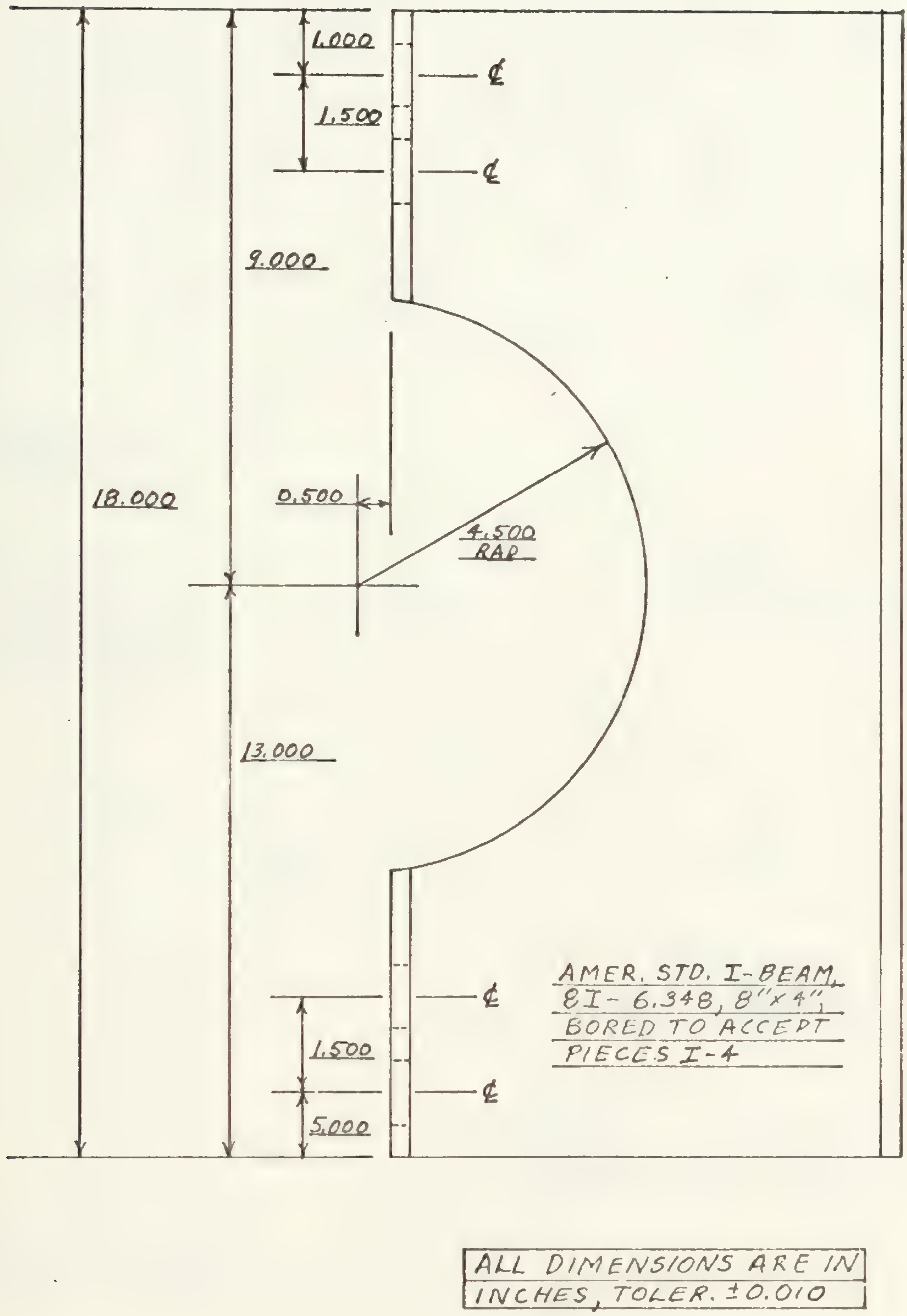


FIGURE V-2. PIECE V-1, PLATFORM SUPPORT

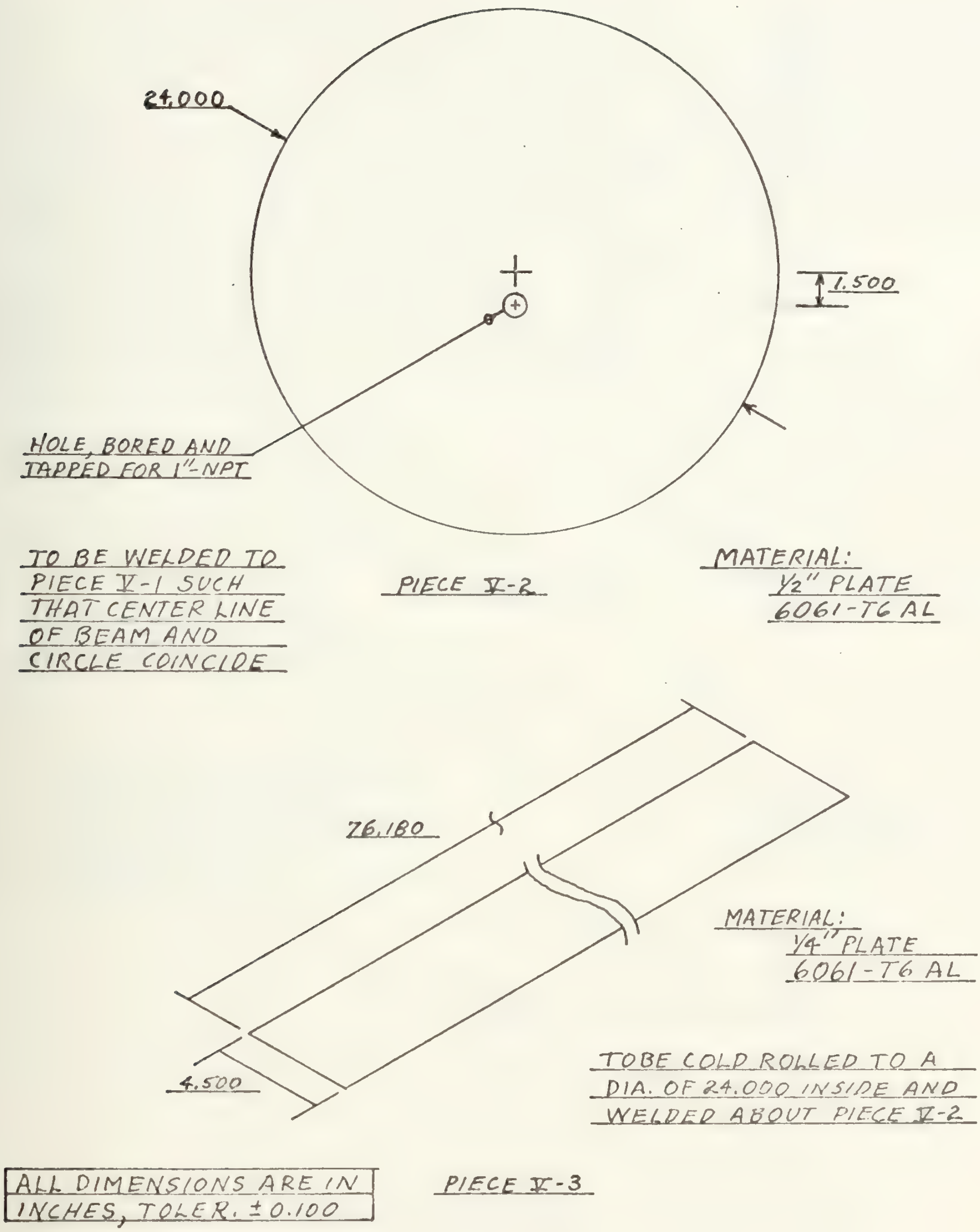


FIGURE V-3. PIECES V-2,3, HOLD DOWN PLATE

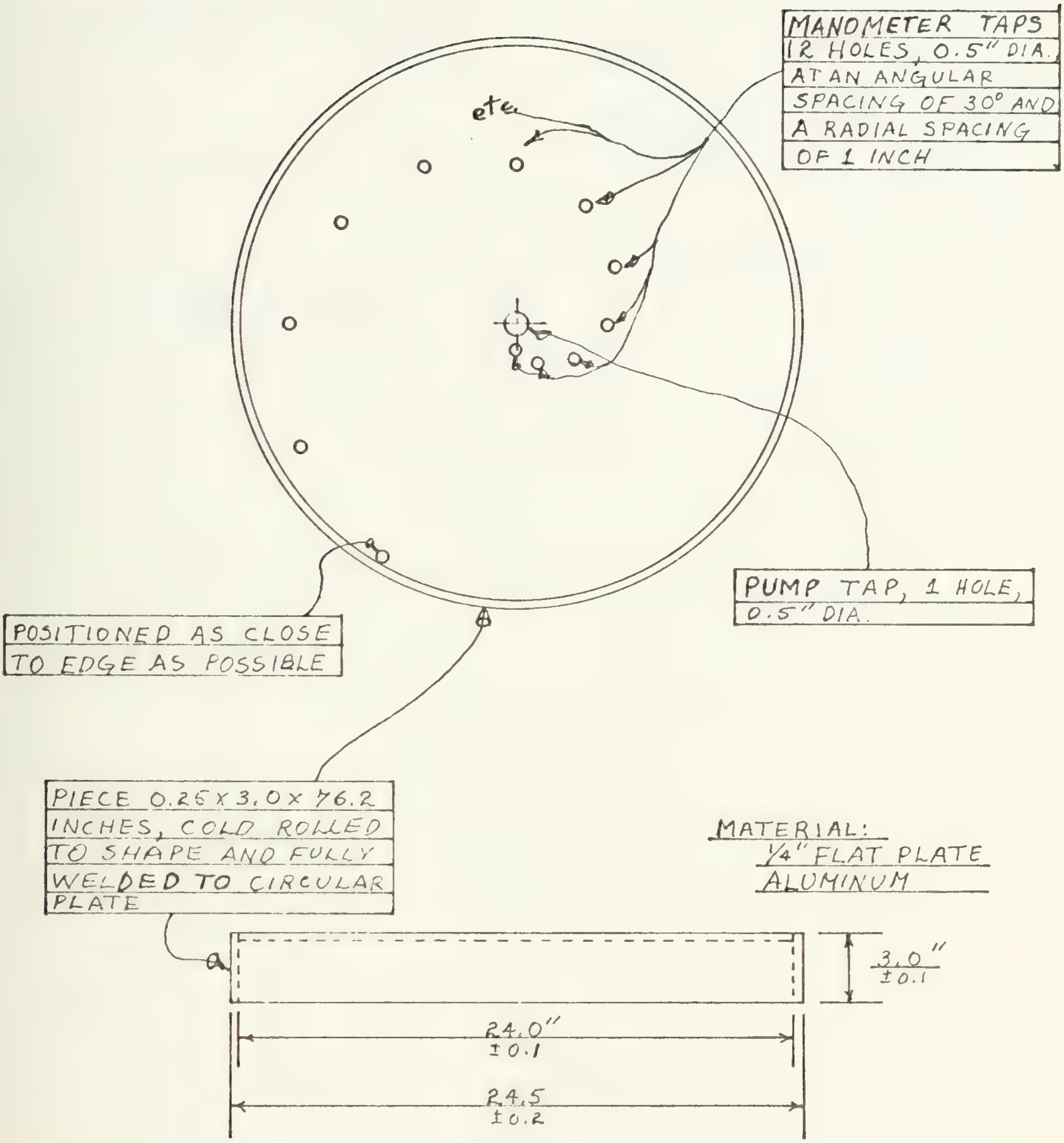


FIGURE V-4. HOLD DOWN TEST PLATE

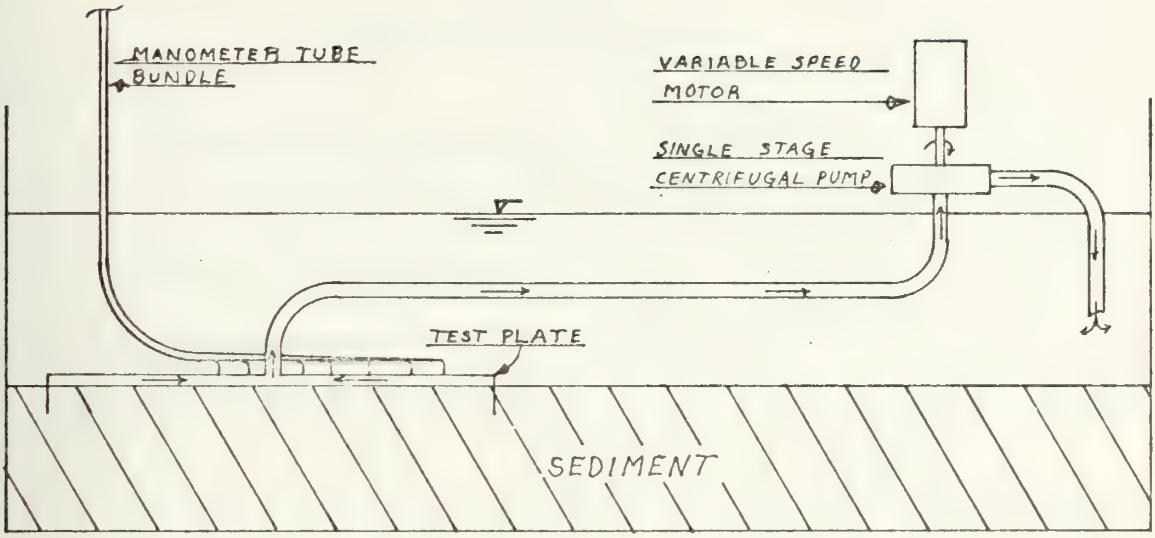


FIGURE V-4-1. CROSS SECTION OF SEDIMENT TANK

APPENDIX C

Compensation Program

The compensation program provides accurate information on center of gravity, center of buoyancy, weight, volume, buoyancy, and density (specific weight) required to compensate the platform. The program will take multiple data sets to provide information on various components during a single execution.

Each data set inputs density, volume, and center of gravity and buoyancy of individual subassemblies relative to a set of coordinates in order to generate the required information of the assembly relative to the same coordinate set. Each data set is totally independent. This allows one to begin with the smallest components, such as bolts, bearings, fabrication pieces, etc., and incrementally build to higher order assemblies; and finally, the platform in total. The output of a data set can be directly used as input in another data set of higher order assemblies. Care must be taken to comply with changes in coordinate systems.

The following is a description of the individual cards of a data set.

Cards 1 - 5	Title cards
Col. 1 - 80	Title of data set, include coordinate locations for reference, FORMAT (20A4)

Card 6	
Col. 1 - 2	Number densities to be used, (N), FORMAT (I2)
Cards 7 - (7+N)	Density Cards
Col. 1 - 10	Density or pseudo density, FORMAT (F10.8)
Col. 11 - 46	Name of material of subassembly, FORMAT (9A4)
Cards (7+N+1) - (last-1)	Subassembly cards
Col. 1 - 36	Name of subassembly, FORMAT (9A4)
Col. 37 - 38	Blank
Col. 39 - 40	Density number, order of density cards determines number, FORMAT (I2)
Col. 41 - 50	Volume displaced by subassembly, FORMAT (F10.4)
Col. 51 - 65	Center of gravity of subassembly, coordinate locations in (x,y,z), FORMAT (3F5.0)
Col. 66 - 80	Center of buoyancy of subassembly, coordinate locations in (x,y,z), FORMAT (3F5.0)
Card (last)	
Col. 1 - 4	Use "GO" if this data set is followed by another intended for execution; use "STOP" if this is the last data set to be executed, FORMAT (A4)

A maximum of 20 Density Cards can be used with this program; a requirement for more than 20 cards can be met by changing the dimension statement. Any number of Subassembly Cards may be used. The execution of these cards will continue until a "STOP" or "GO" card is encountered. Consequently, the first four letters of the name of the subassembly on the Subassembly Cards cannot include "STOP" or "GO". All dimensions of the program are in inches, pounds, and seconds.

```
      DIMENSION DEN(20),NAME(9),X(6),SM(6),NDEN(9),XG(3),XB(3),
     1ITITLE(20)
      DATA IGO/'GO  '/,ISTOP/'STOP'/
  201 WRITE(6,1003)
      SVOL=0.0
      SWT=0.0
      DO 10 I=1,6
   10 SM(I)=0.0
      SDEN=0.037037
      DO 101 I=1,5
      READ(5,1112) (ITITLE(K),K=1,20)
  101 WRITE(6,1001) (ITITLE(K),K=1,20)
      WRITE(6,1002)
      WRITE(6,1000)
      READ(5,1111) I
      DO 100 J=1,I
      READ(5,2222) DEN(J),(NDEN(K),K=1,9)
  100 WRITE(6,2000) J,(NDEN(K),K=1,9),DEN(J)
      WRITE(6,3000)
  500 READ(5,3333) (NAME(J),J=1,9),ID,VOL,(X(J),J=1,6)
      IF(NAME(1).EQ.IGO) GO TO 200
      IF(NAME(1).EQ.ISTOP) GO TO 200
      WT=VOL*DEN(ID)
      SWT=SWT+WT
      SVOL=SVOL+VOL
      DO 300 K=1,3
  300 SM(K)=WT*X(K)+SM(K)
      DO 400 K=4,6
  400 SM(K)=VOL*X(K)+SM(K)
      WRITE(6,4000) (NAME(J),J=1,9),ID,VOL,(X(J),J=1,6)
      GO TO 500
  200 BUOY=SVOL*SDEN-SWT
      DO 600 I=1,3
  600 XG(I)=SM(I)/SWT
      DO 700 I=1,3
  700 XB(I)=SM(I+3)/SVOL
      WRITE(6,4001)
      WRITE(6,5000)(XG(I),I=1,3),(XB(I),I=1,3),BUCY
      PDEN=SWT/SVOL
      WRITE(6,6000) SWT,SVOL,PDEN
      IF(NAME(1).EQ.ISTOP) GO TO 202
      GO TO 201
 1000 FORMAT(/,2X,'DENSITY OF MATERIALS',/,2X,'NO.',6X,'MATERIAL',28X,'D
     1ENSITY')
 1001 FORMAT(10X,20A4)
 1002 FORMAT(//,2X,'*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS *
     1**')
 1003 FORMAT('1')
```

```
2000 FORMAT(2X,I2,2X,9A4,2X,F10.8)
3000 FORMAT(////,4X,'PART NAME',24X,'DEN. NO.',5X,'VOLUME',6X,'CENTER O
    1F GRAVITY',9X,'CENTER OF BUOYANCY',/,61X,'X',8X,'Y',8X,'Z',8X,'X',
    28X,'Y',8X,'Z')
4000 FORMAT(2X,9A4,2X,I2,4X,F10.2,6(3X,F6.2))
4001 FORMAT(//)
5000 FORMAT(2X,'OVERALL CENTER OF GRAVITY',4X,'X=',F6.2,/31X,'Y=',F6.2
    1,/,31X,'Z=',F6.2,/,2X,'OVERALL CENTER OF BUOYANCY',3X,'X=',F6.2,/,
    231X,'Y=',F6.2,/,31X,'Z=',F6.2,/,2X,'BUOYANCY =',F10.2,//)
6000 FORMAT(//,5X,'TOTAL WEIGHT =',F10.2,//,5X,'TOTAL VOLUME =',F10.2
    1,//,5X,'PSEUDO DENSITY =',F10.8,//)
1111 FORMAT(I2)
1112 FORMAT(20A4)
2222 FORMAT(F10.8,9A4)
3333 FORMAT(9A4,2X,I2,F10.4,6F5.0)
 202 WRITE(6,1003)
     STOP
     END
```

The computer output on the following pages is representative of the platform as shown in Plates 1 and 2. The output is shown here only for demonstration purposes and is not intended to be used for the ballasting of a prototype platform. The output does demonstrate the feasibility of the design.

***** INSTRUMENT DRIVE ASSEMBLY, BASIC CENTER OF GRAVITY AND BUOYANCY *****
COORDINATES LOCATED AT THE LOWER, RIGHT, CENTER OF THE FRAME,
BENEATH THE MAIN PLATFORM FRAME. Z IS POSITIVE UPWARD, X IS
POSITIVE TOWARD THE INSTRUMENT, XZ PLANE CONTAINS THE SHAFT
DATE: MAR 10, 1974

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS

NO.	MATERIAL	DENSITY
1	ALUMINUM, 6061-T6	0.09799999
2	STEEL	0.28498000
3	ECCOFLOAT, SERIES PP32	0.01851852
4	DELRIN AF	0.05561100
5	K-MONEL	0.30599999
6	INSULATING OIL	0.03125000
7	AIR AT 3520 PSI	0.01146640
8	TYPICAL SEDIMENT	0.06000000

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	CENTER OF GRAVITY Y	CENTER OF GRAVITY Z	CENTER OF BUOYANCY X	CENTER OF BUOYANCY Y	CENTER OF BUOYANCY Z
UPPER BEARING PLATE	1	112.00	6.57	0.0	63.00	6.57	0.0	63.00
UPPER BEARING PLATE, BOLT INSERTS	4	1.08	5.50	0.0	62.50	5.50	0.0	62.50
UPPER BEARING, PART A	4	0.99	12.00	0.0	63.00	12.00	0.0	63.00
UPPER BEARING, PART B	4	0.40	12.00	0.0	61.90	12.00	0.0	61.90
SCREW SHAFT	2	220.67	12.00	0.0	32.80	12.00	0.0	32.80
UPPER THRUST COLLAR	2	9.87	12.00	0.0	60.20	12.00	0.0	60.20
LOWER THRUST COLLAR	2	9.87	12.00	0.0	3.80	12.00	0.0	3.80
UPPER THRUST BEARING	2	5.44	12.00	0.0	61.20	12.00	0.0	61.20
LOWER THRUST BEARING	2	5.44	12.00	0.0	2.80	12.00	0.0	2.80
LOWER BEARING, PART A	4	0.99	12.00	0.0	1.00	12.00	0.0	1.00
LOWER BEARING, PART B	4	0.40	12.00	0.0	2.10	12.00	0.0	2.10
INSTRUMENT BLOCK	1	203.38	12.65	0.0	56.00	12.65	0.0	56.00
DRIVE CORE OF INST. BLOCK	2	58.02	12.00	0.0	56.00	12.00	0.0	56.00
DRIVE CORE INSULATOR	4	13.20	12.00	0.0	56.00	12.00	0.0	56.00
UPPER MOUNTING BRACKET	1	27.55	-1.10	0.0	18.67	-1.10	0.0	18.67
LOWER MOUNTING BRACKET	1	27.55	-1.10	0.0	5.33	-1.10	0.0	5.33
UPPER MOUNT BRACKET INSERT	4	3.35	-0.78	0.0	19.50	-0.78	0.0	19.50
LOWER MOUNT BRACKET INSERT	4	3.35	-0.78	0.0	3.50	-0.78	0.0	3.50
DRIVE FRAME, PART A	1	314.94	4.05	0.0	32.58	4.05	0.0	32.58
DRIVE FRAME, PART B	1	102.18	6.52	0.0	1.00	6.52	0.0	1.00
DRIVE FRAME, PART C	1	24.49	3.87	0.0	61.50	3.87	0.0	61.50
FLOATATION IN THE FRAME	3	1516.20	4.05	0.0	32.58	4.05	0.0	32.58
BOLTS,3/4,3IN SHANK,LOWER MOUNT BR.,	5	9.53	-0.50	0.0	3.50	-0.50	0.0	3.50
BOLTS,3/4,3IN SHANK,UPPER MOUNT BR.,	5	9.53	-0.50	0.0	18.50	-0.50	0.0	18.50
BOLTS,9/16,3IN,UPPER BEARING PLATE,4	5	4.93	5.50	0.0	61.00	5.50	0.0	61.00
BOLT SPACERS - 1	1	1.46	0.40	0.0	3.50	0.40	0.0	3.50
BOLT SPACERS - 2	1	1.46	0.40	0.0	18.50	0.40	0.0	18.50
BOLT SPACERS - 3	1	2.30	0.40	0.0	54.50	0.40	0.0	54.50

OVERALL CENTER OF GRAVITY X= 8.32
Y= 0.0
Z= 36.31

OVERALL CENTER OF BUOYANCY X= 5.70
Y= 0.0
Z= 34.74

BUOYANCY = -105.33

TOTAL WEIGHT = 204.98

TOTAL VOLUME = 2690.58

PSEUDO DENSITY =0.07618582

```
***** MAIN FRAME ASSEMBLY, BASIC CENTER OF GRAVITY AND BUOYANCY *****
      COORDINATES LOCATED AT THE GEOMETRIC CENTER OF THE LOWER
      FRAME, 6 IN UP FROM FLOOR IF FRAME IS RESTING ON FLANGES
      DATE: MAR 10, 1974
      $$$$$$$

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS
NO.      MATERIAL                              DENSITY
 1    ALUMINUM, 6061-T6                        0.09799999
 2    STEEL                                    0.28498000
 3    ECCOFLOAT, SERIES PP32                   0.01851652
 4    DELRIN  AF                               0.05561100
 5    K-MONEL                                  0.30599999
 6    INSULATING OIL                           0.03125000
 7    AIR AT 3520 PSI                          0.01146640
 8    TYPICAL SEDIMENT                         0.06000000
```

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	Y	Z	CENTER OF BUOYANCY X	Y	Z
LOWER FRAME, AL	1	7360.79	0.0	0.0	0.0	0.0	0.0	0.0
LOWER FRAME, FLOATATION	3	33133.98	0.0	0.0	0.0	0.0	0.0	0.0
SUPPORT COLUMNS, AL	1	2278.65	0.0	0.0	35.99	0.0	0.0	35.99
SUPPORT COLUMNS, FLOATATION	3	13341.95	0.0	0.0	35.99	0.0	0.0	35.99
UPPER PLATFORM, PLATE	1	990.63	0.0	0.0	82.38	0.0	0.0	82.38
UPPER PLATFORM, RING	1	24.98	0.0	0.0	82.75	0.0	0.0	82.75
BOLTS,9/16,1.5IN SHANK,LOWER FRAME	5	55.00	0.0	0.0	0.0	0.0	0.0	0.0
BOLTS,9/16,1.5IN SHANK,SUPPORT COL.	5	55.00	0.0	0.0	32.40	0.0	0.0	32.40

```
OVERALL CENTER OF GRAVITY     X=  0.0
                              Y=  0.0
                              Z= 13.24
OVERALL CENTER OF BUOYANCY    X=  0.0
                              Y=  0.0
                              Z= 11.31
BUOYANCY =     181.52

   TOTAL WEIGHT =   1938.52
   TOTAL VOLUME =  57240.95
   PSEUDO DENSITY =0.03386589
```

**** CORING DEVICE - H BLOWN, NO CORES, CORDINATES LOCATED AT THE BASE OF ****
THE BALLAST TANK, X AXIS POSITIVE IN DIRECTION OF CORING CYLINDER
DRIVE,Z IS ON THE CENTERLINE OF THE TANK, Y IS POSITIVE THRU THE
INSERTION DRIVE.
DATE: MAR 10,1974

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS

NO.	MATERIAL	DENSITY
1	ALUMINUM, 6061-T6	0.09799999
2	STEEL	0.28498000
3	ECCOFLOAT, SERIES PP32	0.01851852
4	DELRIN AF	0.05561100
5	K-MONEL	0.30599999
6	INSULATING OIL	0.03125000
7	AIR AT 3520 PSI	0.01146640
8	TYPICAL SEDIMENT	0.06000000

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	Y	Z	CENTER OF BUOYANCY X	Y	Z
CORING CYLINDER - BODY	1	1747.45	0.0	0.0	42.65	0.0	0.0	42.65
CORING CYLINDER - UPPER BEARING	1	97.19	0.0	0.0	75.40	0.0	0.0	75.40
CORING CYLINDER - LOWER BEARING	1	58.31	0.0	0.0	13.90	0.0	0.0	13.90
CORING CYLINDER - UPPER BEARING	4	23.70	0.0	0.0	74.05	0.0	0.0	74.05
CORING CYLINDER - LOWER BEARING	4	45.89	0.0	0.0	13.90	0.0	0.0	13.90
CORING CYLINDER - FLOATATION	3	49880.69	0.0	0.0	42.65	0.0	0.0	42.65
CORING CYLINDER SUPPORT BEARING L	4	70.37	0.0	0.0	12.20	0.0	0.0	12.20
CORING CYLINDER SUPPORT BEARING U	4	70.37	0.0	0.0	73.10	0.0	0.0	73.10
BALLAST TANK - TOP	1	206.17	0.0	0.0	81.75	0.0	0.0	81.75
BALLAST TANK - CYLINDER	1	3026.92	0.0	0.0	41.00	0.0	0.0	41.00
BALLAST TANK - AIR CONNECTION	1	94.43	0.0	0.0	85.00	0.0	0.0	85.00
BALLAST TANK - CYLINDER SUPPORT	1	449.05	0.0	-0.23	11.75	0.0	-0.23	11.75
BALLAST TANK - CORE GUIDE	1	72.26	0.0	18.75	5.75	0.0	18.75	5.75
BALLAST TANK - FRAME CLAMP	1	465.68	0.0	0.0	5.50	0.0	0.0	5.50
BALLAST TANK - BOLTS,1/2IN	5	10.80	0.0	0.0	5.50	0.0	0.0	5.50
BALLAST TANK HALF BLOWN	7	16930.63	0.0	0.0	61.13	0.0	0.0	61.13
CORE - CYLINDER	2	976.25	0.0	0.0	44.29	0.0	0.0	44.29
CORE - BEARING BLOCK	2	51.24	0.0	0.0	58.34	0.0	0.0	58.34
CORE - LIFTING BEARING ASSM.	2	18.19	0.0	0.0	57.65	0.0	0.0	57.65
CORE - PISTON	2	53.43	0.0	0.0	13.40	0.0	0.0	13.40
CORE - CONNECTOR TO INST. DRIVE	1	312.24	-1.82	23.41	50.87	-1.82	23.41	50.87
CORE - CARRIER	1	129.30	0.0	-1.67	56.15	0.0	-1.67	56.15
CORE - CARRIER SUPPORTS	1	79.88	0.0	0.0	64.43	0.0	0.0	64.43

OVERALL CENTER OF GRAVITY X= -0.03
Y= 0.39
Z= 43.54

OVERALL CENTER OF BUOYANCY X= -0.01
Y= 0.11
Z= 46.57

BUOYANCY = 666.48

TOTAL WEIGHT = 2106.49

TOTAL VOLUME = 74870.19

PSEUDO DENSITY =0.02813517

```
**** CORING DEVICE - H BLOWN,W/CORES  , CORDINATES LOCATED AT THE BASE OF    ****
     THE BALLAST TANK, X AXIS POSITIVE IN DIRECTION OF CORING CYLINDER
     DRIVE,Z IS ON THE CENTERLINE OF THE TANK, Y IS POSITIVE THRU THE
     INSERTION DRIVE.
     DATE: MAR 10,1974
```

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS

NO.	MATERIAL	DENSITY
1	ALUMINUM, 6061-T6	0.09799999
2	STEEL	0.28498000
3	ECCOFLOAT, SERIES PP32	0.01851852
4	DELRIN [illegible]	0.05561100
5	K-MONEL	0.30599999
6	INSULATING OIL	0.03125000
7	AIR AT [illegible]	0.01146640
8	TYPICAL SEDIMENT	0.06000000

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	CENTER OF GRAVITY Y	CENTER OF GRAVITY Z	CENTER OF BUOYANCY X	CENTER OF BUOYANCY Y	CENTER OF BUOYANCY Z
CORING CYLINDER - BODY	1	1747.45	0.0	0.0	42.65	0.0	0.0	42.65
CORING CYLINDER - UPPER BEARING	1	97.19	0.0	0.0	75.40	0.0	0.0	75.40
CORING CYLINDER - LOWER BEARING	1	58.31	0.0	0.0	13.90	0.0	0.0	13.90
CORING CYLINDER - UPPER BEARING	4	23.70	0.0	0.0	74.05	0.0	0.0	74.05
CORING CYLINDER - LOWER BEARING	4	45.89	0.0	0.0	13.90	0.0	0.0	13.90
CORING CYLINDER - FLOATATION	3	49880.69	0.0	0.0	42.65	0.0	0.0	42.65
CORING CYLINDER SUPPORT BEARING L	4	70.37	0.0	0.0	12.20	0.0	0.0	12.20
CORING CYLINDER SUPPORT BEARING U	4	70.37	0.0	0.0	73.10	0.0	0.0	73.10
BALLAST TANK - TOP	1	206.17	0.0	0.0	81.75	0.0	0.0	81.75
BALLAST TANK - CYLINDER	1	3026.92	0.0	0.0	41.00	0.0	0.0	41.00
BALLAST TANK - AIR CONNECTION	1	94.43	0.0	0.0	85.00	0.0	0.0	85.00
BALLAST TANK - CYLINDER SUPPORT	1	449.05	0.0	-0.23	11.75	0.0	-0.23	11.75
BALLAST TANK - CORE GUIDE	1	72.26	0.0	18.75	5.75	0.0	18.75	5.75
BALLAST TANK - FRAME CLAMP	1	465.68	0.0	0.0	5.50	0.0	0.0	5.50
BALLAST TANK - BOLTS,1/2IN	5	10.80	0.0	0.0	5.50	0.0	0.0	5.50
BALLAST TANK HALF BLOWN	7	16930.63	0.0	0.0	61.13	0.0	0.0	61.13
CORE - CYLINDER	2	976.25	0.0	0.0	44.29	0.0	0.0	44.29
CORE - BEARING BLOCK	2	51.24	0.0	0.0	58.34	0.0	0.0	58.34
CORE - LIFTING BEARING ASSM.	2	18.19	0.0	0.0	57.65	0.0	0.0	57.65
CORE PISTON - CORE FULL	2	53.43	0.0	0.0	49.40	0.0	0.0	49.40
CORE - CONNECTOR TO INST. DRIVE	1	312.24	-1.82	23.41	50.87	-1.82	23.41	50.87
CORE - CARRIER	1	129.30	0.0	-1.67	56.15	0.0	-1.67	56.15
CORE - CARRIER SUPPORTS	1	79.88	0.0	0.0	64.43	0.0	0.0	64.43
CORES FULL - 3 FT	8	1413.72	0.0	0.0	30.40	0.0	0.0	30.40

```
OVERALL CENTER OF GRAVITY     X= -0.03
                              Y=  0.37
                              Z= 43.28
OVERALL CENTER OF BUOYANCY    X= -0.01
                              Y=  0.11
                              Z= 46.30
BUOYANCY =    634.02
```

TOTAL WEIGHT =   2191.31

TOTAL VOLUME = 76283.88

PSEUDO DENSITY =0.02872571

**** CORING DEVICE - BLOWN , NO CORES, CORDINATES LOCATED AT THE BASE OF ****
THE BALLAST TANK, X AXIS POSITIVE IN DIRECTION OF CORING CYLINDER
DRIVE,Z IS ON THE CENTERLINE OF THE TANK, Y IS POSITIVE THRU THE
INSERTION DRIVE.
DATE: MAR 10,1974

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS

NO.	MATERIAL	DENSITY
1	ALUMINUM, 6061-T6	0.09799999
2	STEEL	0.28498000
3	ECCOFLOAT, SERIES PP32	0.01851852
4	DELRIN AF	0.05561100
5	K-MONEL	0.30599999
6	INSULATING OIL	0.03125000
7	AIR AT 3520 PSI	0.01146640
8	TYPICAL SEDIMENT	0.06000000

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	Y	Z	CENTER OF BUOYANCY X	Y	Z
CORING CYLINDER - BODY	1	1747.45	0.0	0.0	42.65	0.0	0.0	42.65
CORING CYLINDER - UPPER BEARING	1	97.19	0.0	0.0	75.40	0.0	0.0	75.40
CORING CYLINDER - LOWER BEARING	1	58.31	0.0	0.0	13.90	0.0	0.0	13.90
CORING CYLINDER - UPPER BEARING	4	23.70	0.0	0.0	74.05	0.0	0.0	74.05
CORING CYLINDER - LOWER BEARING	4	45.89	0.0	0.0	13.90	0.0	0.0	13.90
CORING CYLINDER - FLOATATION	3	49880.69	0.0	0.0	42.65	0.0	0.0	42.65
CORING CYLINDER SUPPORT BEARING L	4	70.37	0.0	0.0	12.20	0.0	0.0	12.20
CORING CYLINDER SUPPORT BEARING U	4	70.37	0.0	0.0	73.10	0.0	0.0	73.10
BALLAST TANK - TOP	1	206.17	0.0	0.0	81.75	0.0	0.0	81.75
BALLAST TANK - CYLINDER	1	3026.92	0.0	0.0	41.00	0.0	0.0	41.00
BALLAST TANK - AIR CONNECTION	1	94.43	0.0	0.0	85.00	0.0	0.0	85.00
BALLAST TANK - CYLINDER SUPPORT	1	449.05	0.0	-0.23	11.75	0.0	-0.23	11.75
BALLAST TANK - CORE GUIDE	1	72.26	0.0	18.75	5.75	0.0	18.75	5.75
BALLAST TANK - FRAME CLAMP	1	465.68	0.0	0.0	5.50	0.0	0.0	5.50
BALLAST TANK - BOLTS,1/2IN	5	10.80	0.0	0.0	5.50	0.0	0.0	5.50
BALLAST TANK BLOWN	7	33861.26	0.0	0.0	40.75	0.0	0.0	40.75
CORE - CYLINDER	2	976.25	0.0	0.0	44.29	0.0	0.0	44.29
CORE - BEARING BLOCK	2	51.24	0.0	0.0	58.34	0.0	0.0	58.34
CORE - LIFTING BEARING ASSM.	2	18.19	0.0	0.0	57.65	0.0	0.0	57.65
CORE - PISTON	2	53.43	0.0	0.0	13.40	0.0	0.0	13.40
CORE - CONNECTOR TO INST. DRIVE	1	312.24	-1.82	23.41	50.87	-1.82	23.41	50.87
CORE - CARRIER	1	129.30	0.0	-1.67	56.15	0.0	-1.67	56.15
CORE - CARRIER SUPPORTS	1	79.88	0.0	0.0	64.43	0.0	0.0	64.43

OVERALL CENTER OF GRAVITY X= -0.02
Y= 0.36
Z= 41.56

OVERALL CENTER OF BUOYANCY X= -0.01
Y= 0.09
Z= 41.74

BUOYANCY = 1099.41

TOTAL WEIGHT = 2300.62

TOTAL VOLUME = 91800.81

PSEUDO DENSITY =0.02506099

**** CORING DEVICE - BLOWN ,W/CORES , CORDINATES LOCATED AT THE BASE OF ****
THE BALLAST TANK, X AXIS POSITIVE IN DIRECTION OF CORING CYLINDER
DRIVE,Z IS ON THE CENTERLINE OF THE TANK, Y IS POSITIVE THRU THE
INSERTION DRIVE.
DATE: MAR 10,1974

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS

NO.	MATERIAL	DENSITY
1	ALUMINUM, 6061-T6	0.09799999
2	STEEL	0.28498000
3	ECCOFLOAT, SERIES PP32	0.01851852
4	DELRIN AF	0.05561100
5	K-MONEL	0.30599999
6	INSULATING OIL	0.03125000
7	AIR AT 3520 PSI	0.01146640
8	TYPICAL SEDIMENT	0.06000000

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	Y	Z	CENTER OF BUOYANCY X	Y	Z
CORING CYLINDER - BODY	1	1747.45	0.0	0.0	42.65	0.0	0.0	42.65
CORING CYLINDER - UPPER BEARING	1	97.19	0.0	0.0	75.40	0.0	0.0	75.40
CORING CYLINDER - LOWER BEARING	1	58.31	0.0	0.0	13.90	0.0	0.0	13.90
CORING CYLINDER - UPPER BEARING	4	23.70	0.0	0.0	74.05	0.0	0.0	74.05
CORING CYLINDER - LOWER BEARING	4	45.89	0.0	0.0	13.90	0.0	0.0	13.90
CORING CYLINDER - FLOATATION	3	49880.69	0.0	0.0	42.65	0.0	0.0	42.65
CORING CYLINDER SUPPORT BEARING L	4	70.37	0.0	0.0	12.20	0.0	0.0	12.20
CORING CYLINDER SUPPORT BEARING U	4	70.37	0.0	0.0	73.10	0.0	0.0	73.10
BALLAST TANK - TOP	1	206.17	0.0	0.0	81.75	0.0	0.0	81.75
BALLAST TANK - CYLINDER	1	3026.92	0.0	0.0	41.00	0.0	0.0	41.00
BALLAST TANK - AIR CONNECTION	1	94.43	0.0	0.0	85.00	0.0	0.0	85.00
BALLAST TANK - CYLINDER SUPPORT	1	449.05	0.0	-0.23	11.75	0.0	-0.23	11.75
BALLAST TANK - CORE GUIDE	1	72.26	0.0	18.75	5.75	0.0	18.75	5.75
BALLAST TANK - FRAME CLAMP	1	465.68	0.0	0.0	5.50	0.0	0.0	5.50
BALLAST TANK - BOLTS,1/2IN	5	10.80	0.0	0.0	5.50	0.0	0.0	5.50
BALLAST TANK BLOWN	7	35861.26	0.0	0.0	40.75	0.0	0.0	40.75
CORE - CYLINDER	2	976.25	0.0	0.0	44.29	0.0	0.0	44.29
CORE - BEARING BLOCK	2	51.24	0.0	0.0	58.34	0.0	0.0	58.34
CORE - LIFTING BEARING ASSM.	2	18.19	0.0	0.0	57.65	0.0	0.0	57.65
CORE PISTON - CORE FULL	2	53.43	0.0	0.0	49.40	0.0	0.0	49.40
CORE - CONNECTOR TO INST. DRIVE	1	312.24	-1.82	23.41	50.87	-1.82	23.41	50.87
CORE - CARRIER	1	129.30	0.0	-1.67	56.15	0.0	-1.67	56.15
CORE - CARRIER SUPPORTS	1	79.88	0.0	0.0	64.43	0.0	0.0	64.43
CORES FULL - 3 FT	8	1413.72	0.0	0.0	30.40	0.0	0.0	30.40

OVERALL CENTER OF GRAVITY X= -0.02
Y= 0.34
Z= 41.41

OVERALL CENTER OF BUOYANCY X= -0.01
Y= 0.09
Z= 41.59

BUOYANCY = 1066.94

TOTAL WEIGHT = 2385.44

TOTAL VOLUME = 93214.50

PSEUDO DENSITY =0.02559089

```
**** INSTRUMENT DRIVE MOTOR BOX AND DRIVE, COORDINATES LOCATED AT THE LOWER ****
     RIGHT OF FIG IV-1-1, X IS POSITIVE ALONG THE BOTTOM, Z IS POSITIVE
     UPWARD, ALL SHAFTS LIE IN THE XZ PLANE, Y IS POSITIVE TOWARD THE
     COVER
     DATE: MAR 15,1974

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***
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DENSITY OF MATERIALS

NO.	MATERIAL	DENSITY
1	ALUMINUM, 6061-T6	0.09799999
2	STEEL	0.28498000
3	ECCOFLOAT, SERIES PP32	0.01851852
4	DELRIN AF	0.05561100
5	K-MONEL	0.30599999
6	INSULATING OIL	0.03125000
7	AIR AT 3520 PSI	0.01146640
8	TYPICAL SEDIMENT	0.06000000

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	Y	Z	CENTER OF BUOYANCY X	Y	Z
PINION GEAR	2	1.40	9.17	0.0	16.50	9.17	0.0	16.50
BULL GEAR	2	23.16	4.50	0.0	17.00	4.50	0.0	17.00
SHAFT, 7/8 IN, 10.0 IN LONG	2	6.01	4.50	0.0	19.25	4.50	0.0	19.25
BEARINGS	2	10.53	4.50	0.0	16.75	4.50	0.0	16.75
PIECE IV-1-1, MOTOR MOUNT PLATE	1	21.33	5.88	0.0	15.38	5.88	0.0	15.38
MOTOR, 1/2 HP	2	115.80	9.17	0.0	10.47	9.17	0.0	10.47
INSULATING OIL	6	418.07	6.50	0.0	18.00	6.50	0.0	18.00
FLOATATION , INTERNAL	3	1600.00	4.50	0.0	7.00	4.50	0.0	7.00
TOP PLATE	1	26.75	6.41	0.0	19.88	6.41	0.0	19.88
PRESSURE COMPENSATION CONNECTION	1	2.75	10.75	0.0	20.75	10.75	0.0	20.75
SIDE, MOUNT	1	44.56	0.13	0.0	9.99	0.13	0.0	9.99
SIDE	1	45.00	12.88	0.0	10.00	12.88	0.0	10.00
BOTTOM	1	27.68	6.55	0.0	0.13	6.55	0.0	0.13
ELECTRICAL LEAD CONNECTION	1	2.21	3.25	0.0	-1.00	3.25	0.0	-1.00
BACK	1	65.00	6.50	-4.63	10.00	6.50	-4.63	10.00
COVER	1	76.93	6.50	4.63	10.00	6.50	4.63	10.00
RETAINERS	1	16.46	6.50	4.25	10.00	6.50	4.25	10.00
TORQUE LIMITER, 25 TEETH, 3/8 PITCH	2	4.91	4.50	0.0	22.00	4.50	0.0	22.00
SPROCKET, 35 TEETH, 3/8 IN PITCH	2	4.91	-12.00	0.0	22.00	-12.00	0.0	22.00
CHAIN, 108 PITCHES, STAINLESS STEEL	2	2.37	-3.75	0.0	22.00	-3.75	0.0	22.00

```
OVERALL CENTER OF GRAVITY      X=  6.26
                               Y=  0.10
                               Z= 11.37
OVERALL CENTER OF BUOYANCY     X=  5.27
                               Y=  0.05
                               Z=  9.66
BUOYANCY =    -29.91

   TOTAL WEIGHT =     123.09

   TOTAL VOLUME =    2515.84

   PSEUDO DENSITY =0.04892755
```

**** CORING CYLINDER MOTOR BOX AND DRIVE, COORDINATES ARE LOCATED AT THE ****
BOTTOM RIGHT OF FIG IV-2, X AXIS IS POSITIVE ALONG THE BOTTOM, Z AXIS
IS POSITIVE UPWARD, ALL SHAFTS LIE IN THE XZ PLANE, Y IS POSITIVE
TOWARD THE COVER.
DATE: MAR 11, 1974

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS

NO.	MATERIAL	DENSITY
1	ALUMINUM, 6061-T6	0.09799999
2	STEEL	0.28498000
3	ECCOFLOAT, SERIES PP32	0.01851852
4	DELRIN AF	0.05561100
5	K-MONEL	0.30599999
6	INSULATING OIL	0.03125000
7	AIR AT 3520 PSI	0.01140670
8	TYPICAL SEDIMENT	0.06000000

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	Y	Z	CENTER OF BUOYANCY X	Y	Z
GEARS	2	43.93	6.31	0.0	15.50	6.31	0.0	15.50
BEARINGS	2	22.46	6.31	0.0	17.75	6.31	0.0	17.75
MOTOR, 1/12 HP	2	42.11	4.13	0.0	6.44	4.13	0.0	6.44
SHAFT 1, 7.5 IN	2	2.30	8.50	0.0	11.50	8.50	0.0	11.50
SHAFT 2 , 7.5 IN	2	2.30	4.13	0.0	15.50	4.13	0.0	15.50
SHAFT 3 , 7.5 IN	2	2.30	8.50	0.0	19.50	8.50	0.0	19.50
SHAFT 4 , 16.0 IN	2	4.91	4.13	0.0	25.75	4.13	0.0	25.75
PIECES IV-2-2, BEARING MOUNT PLATES	1	80.18	6.50	0.0	17.38	6.50	0.0	17.38
PIECE IV-2-3, MOTOR MOUNT PLATE	1	22.95	7.50	0.0	9.38	7.50	0.0	9.38
INSULATING OIL	6	1801.57	6.50	0.0	17.90	6.50	0.0	17.90
FLOATATION , INTERNAL	3	900.00	9.00	0.0	4.50	9.00	0.0	4.50
BOTTOM PLATE	1	28.13	6.50	0.0	0.13	6.50	0.0	0.13
SIDE PLATE	1	59.63	0.13	0.0	13.25	0.13	0.0	13.25
SIDE PLATE	1	59.18	12.88	0.0	13.32	12.88	0.0	13.32
ELECTRICAL LEAD CONNECTION	1	2.21	13.88	0.0	4.00	13.88	0.0	4.00
TOP PLATE	1	26.59	6.30	0.0	26.38	6.30	0.0	26.38
PRESSURE COMPENSATION	1	2.75	8.50	0.0	27.50	8.50	0.0	27.50
BACK PLATE	1	86.13	6.50	-4.63	13.25	6.50	-4.63	13.25
FRONT COVER	1	97.63	6.50	4.63	12.88	6.50	4.63	12.88
RETAINERS, EQUAL ANGLE	1	18.96	6.50	4.13	12.88	6.50	4.13	12.88
SPROCKET, 12 TEETH, 1 IN PITCH	2	11.40	4.13	0.0	30.90	4.13	0.0	30.90
SPROCKET, 96 TEETH, 1 IN PITCH	2	140.45	4.13	-29.80	30.90	4.13	-29.80	30.90
CHAIN, 120 PITCHES, STAINLESS STEEL	2	56.14	4.13	-25.70	30.90	4.13	-25.70	30.90
CAM, SHUT OFF	2	22.23	4.13	0.0	28.50	4.13	0.0	28.50
CONNECTING PLATE	1	86.42	6.50	-3.42	26.75	6.50	-3.42	26.75
BOLTS, 1/2 IN, 1.5 IN SHANK	5	5.10	6.50	-3.87	26.75	6.50	-3.87	26.75

OVERALL CENTER OF GRAVITY X= 5.87
Y= -7.06
Z= 19.38

OVERALL CENTER OF BUOYANCY X= 6.95
Y= -1.60
Z= 14.89

BUOYANCY = -95.99

TOTAL WEIGHT = 230.36

TOTAL VOLUME = 3627.95

PSEUDO DENSITY =0.06349474

```
**** PLATFORM SUPPORT - HOLD DOWN PLATE - COORDINATES ARE LOCATED AT THE     ****
     BOTTOM OF THE SKIRT - Z AXIS IS POSITIVE THROUGH THE CENTER OF THE
     PLATE AND SUPPORT - X AXIS IS POSITIVE TOWARD THE MAIN FRAME -
     Y AXIS IS PARALLEL TO THE PIPE IN THE MAIN FRAME
     DATE: APR 10,1974

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS
NO.     MATERIAL                                  DENSITY
 1   ALUMINUM, 6061-T6                          0.09799999
 2   STEEL                                      0.28498000
 3   ECCOFLOAT, SERIES PP32                     0.01851852
 4   DELRIN  AF                                 0.05561100
 5   K-MONEL                                    0.30599999
 6   INSULATING OIL                             0.03125000
 7   AIR AT 3520 PSI                            0.01146640
 8   TYPICAL SEDIMENT                           0.06000000
```

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	CENTER OF GRAVITY Y	CENTER OF GRAVITY Z	CENTER OF BUOYANCY X	CENTER OF BUOYANCY Y	CENTER OF BUOYANCY Z
I - BEAM	1	103.65	-1.00	0.0	15.50	-1.00	0.0	15.50
UPPER MOUNT BRACKET	1	27.55	5.00	0.0	23.75	5.00	0.0	23.75
LOWER MOUNT BRACKET	1	27.55	5.00	0.0	7.25	5.00	0.0	7.25
UPPER MOUNT BRACKET INSERTS	4	3.35	5.00	0.0	24.75	5.00	0.0	24.75
LOWER MOUNT BRACKET INSERTS	4	3.35	5.00	0.0	6.25	5.00	0.0	6.25
BOLTS,3/4,3IN SHANK, UPPER MOUNT BR.	5	9.53	5.00	0.0	24.75	5.00	0.0	24.75
BOLTS,3/4,3IN SHANK, LOWER MOUNT BR.	5	9.53	5.00	0.0	6.25	5.00	0.0	6.25
PLATE, HOLD DOWN	1	113.10	0.0	0.0	4.38	0.0	0.0	4.38
SKIRT	1	85.70	0.0	0.0	2.25	0.0	0.0	2.25

```
OVERALL CENTER OF GRAVITY     X=  1.16
                              Y=  0.0
                              Z=  9.81
OVERALL CENTER OF BUOYANCY    X=  0.78
                              Y=  0.0
                              Z=  9.26
BUOYANCY =     -27.05


   TOTAL WEIGHT =      41.24

   TOTAL VOLUME =     383.32

   PSEUDO DENSITY =0.10759950
```

**** --- PLATFORM --- COORDINATES ARE LOCATED AT THE BASE OF THE BALLAST ****
TANK - Z AXIS IS POSITIVE ALONG THE CENTERLINE OF THE TANK - Y AXIS
IS POSITIVE TOWARD THE CORE INSERTION DRIVE - X AXIS IS POSITIVE AWAY
FROM THE CORE CYLINDER DRIVE
DATE: APR 16,1974

*** ALL DIMENSIONS ARE INCHES, POUNDS, AND SECONDS ***

DENSITY OF MATERIALS

NO.	MATERIAL	DENSITY
1	MAIN FRAME	0.03386589
2	CORING DEVICE, HALF BLOWN, W/CORES	0.02872571
3	CORE CYL. MOTOR BOX AND DRIVE	0.06349468
4	INSTRUMENT DRIVE	0.07618576
5	INSTRUMENT DRIVE MOTOR BOX	0.04892755
6	PLATFORM SUPPORT - HOLD DOWN PLATE	0.10759950
7	CONTROL BOX	0.04415500
8	BATTERY BOX	0.06617576
9	AIR REGULATORS AND PIPING	0.28484797
10	AIR FLASKS AND AIR AT 8000PSI	0.06249418
11	MISC.	0.28498000
12	6061-T6 ALUMINUM	0.09799999
13	ECCOFLOAT, SERIES PP32	0.01851852

PART NAME	DEN. NO.	VOLUME	CENTER OF GRAVITY X	Y	Z	CENTER OF BUOYANCY X	Y	Z
MAIN FRAME	1	57240.95	0.0	0.0	19.24	0.0	0.0	17.31
CORING DEVICE, HALF BLOWN, W/CORES	2	76263.88	0.03	0.37	43.28	0.01	0.11	46.30
CORE CYL. MOTOR BOX AND DRIVE	3	3627.95	0.60	-1.75	66.18	-4.90	-2.82	61.69
INSTRUMENT DRIVE NO.1	4	2690.58	-36.00	62.32	30.31	-36.00	59.70	28.74
INSTRUMENT DRIVE NO.2	4	2690.58	0.0	62.32	30.31	0.0	59.70	28.74
INSTRUMENT DRIVE NO.3	4	2690.58	36.00	62.32	30.31	36.00	59.70	28.74
INSTRUMENT DRIVE NO.4	4	2690.58	62.32	36.00	30.31	59.70	36.00	28.74
INSTRUMENT DRIVE NO.5	4	2690.58	62.32	-36.00	30.31	59.70	-36.00	28.74
INSTRUMENT DRIVE NO.6	4	2690.58	18.00	-62.30	30.31	18.00	-59.70	28.74
INSTRUMENT DRIVE NO.7	4	2690.58	-18.00	-62.30	30.31	-18.00	-59.70	28.74
INSTRUMENT DRIVE NO.8	4	2690.58	-62.30	0.0	30.31	-59.70	0.0	28.74
INSTRUMENT DRIVE MOTOR BOX NO. 1	5	2515.84	-36.10	47.74	47.37	-36.10	48.73	45.66
INSTRUMENT DRIVE MOTOR BOX NO. 2	5	2515.84	-0.10	47.74	47.37	-0.05	48.73	45.66
INSTRUMENT DRIVE MOTOR BOX NO. 3	5	2515.84	35.90	47.74	47.37	35.95	48.73	45.66
INSTRUMENT DRIVE MOTOR BOX NO. 4	5	2515.84	47.74	36.10	47.37	48.73	36.05	45.66
INSTRUMENT DRIVE MOTOR BOX NO. 5	5	2515.84	47.74	-35.90	47.37	48.73	-35.90	45.66
INSTRUMENT DRIVE MOTOR BOX NO. 6	5	2515.84	18.10	-47.70	47.37	18.05	-48.70	45.66
INSTRUMENT DRIVE MOTOR BOX NO. 7	5	2515.84	-17.90	-47.70	47.37	-17.90	-48.70	45.66
INSTRUMENT DRIVE MOTOR BOX NO. 8	5	2515.84	-47.70	-0.10	47.37	-48.70	-0.05	45.66
INSTRUMENT DRIVE FOR CORING	4	2690.58	7.68	25.00	30.31	10.30	25.00	28.74
INST. DR. FOR CORING, MOTOR BOX	5	2515.84	22.26	25.10	47.37	21.27	25.05	45.66
HOLD DOWN PLATE NO. 1	6	383.32	46.00	56.84	-7.69	46.00	57.22	-8.24
HOLD DOWN PLATE NO. 2	6	383.32	56.84	-46.00	-7.69	57.22	46.00	-8.24
HOLD DOWN PLATE NO. 3	6	383.32	-46.00	-56.80	-7.69	-46.00	-57.20	-8.24
HOLD DOWN PLATE NO. 4	6	383.32	-56.80	46.00	-7.69	-57.20	46.00	-8.24
CONTROL BOX	7	3456.00	31.00	0.0	16.00	31.00	0.0	16.00
BATTERY BOX	8	8964.00	0.0	-31.00	18.00	0.0	-31.00	18.00
AIR REGULATORS AND PIPING	9	363.00	-30.50	19.00	13.00	-30.50	19.00	13.00
AIR FLASKS AND AIR AT 8000PSI	10	51576.00	0.0	0.0	42.00	0.0	0.0	42.00
MISC. (CABLES,TUBING,ETC.)	11	350.90	0.0	0.0	20.00	0.0	0.0	20.00
LIFTING HARNESS	11	160.20	0.0	0.0	90.00	0.0	0.0	90.00
MAIN FLOAT, AL	12	2091.86	0.0	0.0	158.00	0.0	0.0	158.00
FLOATATION, MAIN FLOAT,6-3FT,12-4FT	13	137978.69	0.0	0.0	163.00	0.0	0.0	163.00

OVERALL CENTER OF GRAVITY X= 1.71
Y= 0.75
Z= 59.44

OVERALL CENTER OF BUOYANCY X= 1.14
Y= 0.45
Z= 81.17

BUOYANCY = 6.34

TOTAL WEIGHT = 14456.02

TOTAL VOLUME = 390483.54

PSEUDO DENSITY =0.03702077

APPENDIX D

Operation of the Platform

The following is a narrative of a typical operation cycle using the platform. The support vessel is assumed to be similar to the MAXINE D which is currently used to support the Autec vehicles SEA CLIFF and TURTLE. SOPDOSS has been totally readied in port and has been tested. SOPDOSS is located next to the submersible on the fantail of the support ship beneath the boom of a 50 ton crane.

Upon reaching the assigned operating area, the main float of SOPDOSS is attached to the main hoist of the crane and lifted to a position above SOPDOSS. The four legged lifting harness is attached to the base of the main float and connected to the main frame of SOPDOSS. SOPDOSS is unlashed from the deck and the crane rapidly raises the platform, swings it over the side, and begins to lower the platform into the sea. Since SOPDOSS is positively buoyant with the ballast tank empty, the upper portion of the main float will be awash as SOPDOSS floats on the surface. The crane cable is detached by divers and the 10 ton lowering boom cable is attached to the main float with the disconnect to be used by the submersible. When all is ready for lowering, a diver proceeds to the top of the platform and vents the ballast tank until SOPDOSS becomes negatively buoyant and takes up the slack in the lowering cable. The vent valve is manually operated by pulling a lanyard on the override lever. Care is

taken not to vent all the air. Any captured air will be compressed as SOPDOSS proceeds downward and will reduce the amount of air required to bring SOPDOSS to neutral buoyancy.

When the diver is clear of SOPDOSS, the platform is lowered to the ocean floor. The cable tension will increase as cable is payed out due to the weight of the cable and increasing negative buoyancy of SOPDOSS due to the compression of the air in the ballast tank. Active sonar is used to track SOPDOSS during its descent. As SOPDOSS approaches the floor the rate of cable payout is decreased and the platform is landed.

During the lowering of SOPDOSS, the submersible has been launched and is proceeding to rendezvous with SOPDOSS. A small, battery powered transponder on SOPDOSS assists the submersible in location. Upon location of SOPDOSS, the lowering cable is disconnected and the submersible signals the support ship to haul in the cable.

The submersible then locates the umbilical cable and attaches itself to the platform. The manipulators are lowered and extended forward to grasp the top plate of the main frame of the platform. The blow valve is then pulsed to blow the ballast tank. When the platform just reaches neutral buoyancy, SOPDOSS is maneuvered off the ocean floor and is transported to the first site for testing. At the site, the platform vent valve is pulsed and SOPDOSS settles to the floor. The hold down pumps are started and the instruments are individually inserted into the sediment.

When the instruments are ready, the hold down pumps are secured and testing is begun. During this time, the submersible can release the platform and maneuver about the platform to observe the operations. Upon completion of testing, the hold down pumps are started and a core of the sediment is taken and the core cylinder is rotated to bring a new core barrel into position. The hold down pumps are then secured and the submersible again grasps the platform. The hold down pumps are momentarily reversed and the blow valve is pulsed to bring the platform back to neutral buoyancy. The platform is then transported to a new site. Care is taken not to raise the platform too far above the floor since the air in the ballast tank will expand and cause the platform to become positively buoyant. Since it is possible for this process to get out of hand, it is the most dangerous portion of the operation and must be carefully accomplished. If the sites are at different depths, the ballasting system may have to be operated during transit.

Upon completion of all testing, the hold down pumps are reversed to release the platform from the bottom and then secured. The submersible disconnects the umbilical cable and contacts the support ship to ensure that the surface above the platform is clear of shipping. When the surface is clear, the submersible approaches the platform and locates the lanyard attached to the manual override on the blow valve. Standing clear of the platform, the submersible operates the blow valve bringing the platform to positive buoyancy. As

the platform rises, the air in the ballast tank expands eventually blowing the tank dry. The submersible then proceeds to the surface.

The support ship locates SOPDOSS on the surface and proceeds with the pickup of the platform and submersible. On board the support ship, the core barrels are removed from SOPDOSS and the platform and submersible are thoroughly washed down with fresh water. The batteries and air flasks can then be charged if necessary for continued operation.

Approximately 16 hours should be allotted for a complete operational cycle, 8 hours of which would be operations on the ocean floor.

APPENDIX E

Material Costs

The estimated material costs as presented in Table 1 are an accurate representation of such costs during April, 1974. The present state of the economy demands that these figures be used only as a guideline for future purchasing. The prices were obtained from sales representatives of the various manufacturers.

The cost of the Eccofloat may be reduced by 10-20% if purchased by contract with long lead time. The aluminum and steel are readily available from most commercial outlets. The 24 x 1/2 in. pipe used for the ballast tank will have to be special ordered from Alcoa. If a long lead time is allowed, the cost of this pipe will be near that of structural aluminum.

Currently there are no K-Monel bolts available for purchase. Bolts of 1040 steel, grade 5, have been substituted. These bolts have the same strength as K-Monel, but are easily corroded and will have to be checked and changed often during operation of the platform.

The air flasks are manufactured per order by National Tube Division, United States Steel. Again, a long lead time is desirable in terms of costs. They also manufacture the high pressure piping required.

All other items are from normal stock and there should be no problem with acquisition.

TABLE 1, MATERIAL COSTS

1.	Eccofloat, Series PP32, 5672 lb	$ 19,500.00
2.	6061-T6 aluminum, 2890 lb	2,890.00
3.	1020 structural steel, 1130 lb	282.00
4.	Delrin AF, stock	80.00
5.	Nylon screw insulators, 1000 ea., assorted	12.00
6.	Nylon hex bolts, 1000 ea.	22.00
7.	Bolts and nuts, 1040 steel, grade 5, assorted	300.00
8.	Batteries, 8 ea.	285.00
9.	Power relays, 15 ea.	2,000.00
10.	Circuit breaker	230.00
11.	Limit switches, 19 ea.	25.00
12.	Solenoid valves, 2 ea.	68.00
13.	Air regulators, 5 ea.	1,126.00
14.	Relief valves, 5 ea.	377.00
15.	Pressure compensators, 20 ea.	1,123.00
16.	Battery charging connection	100.00
17.	Umbilical cable and connections	410.00
18.	3/4 HP motors, 9 ea.	1,170.00
19.	1/8 HP motors, 4 ea.	480.00
20.	1/12 HP motor	80.00
21.	Vane pumps, 4 ea.	120.00
22.	Gears, 26 ea., various sizes	320.00
23.	Flange bearings, 26 ea.	260.00
24.	Sprockets and chain	250.00
25.	Torque limiters, 10 ea.	359.00
26.	Air flasks, 2.5 ft^3, 8 ea.	12,000.00
27.	High pressure piping and fittings	500.00
	TOTAL:	$ 44,369.00

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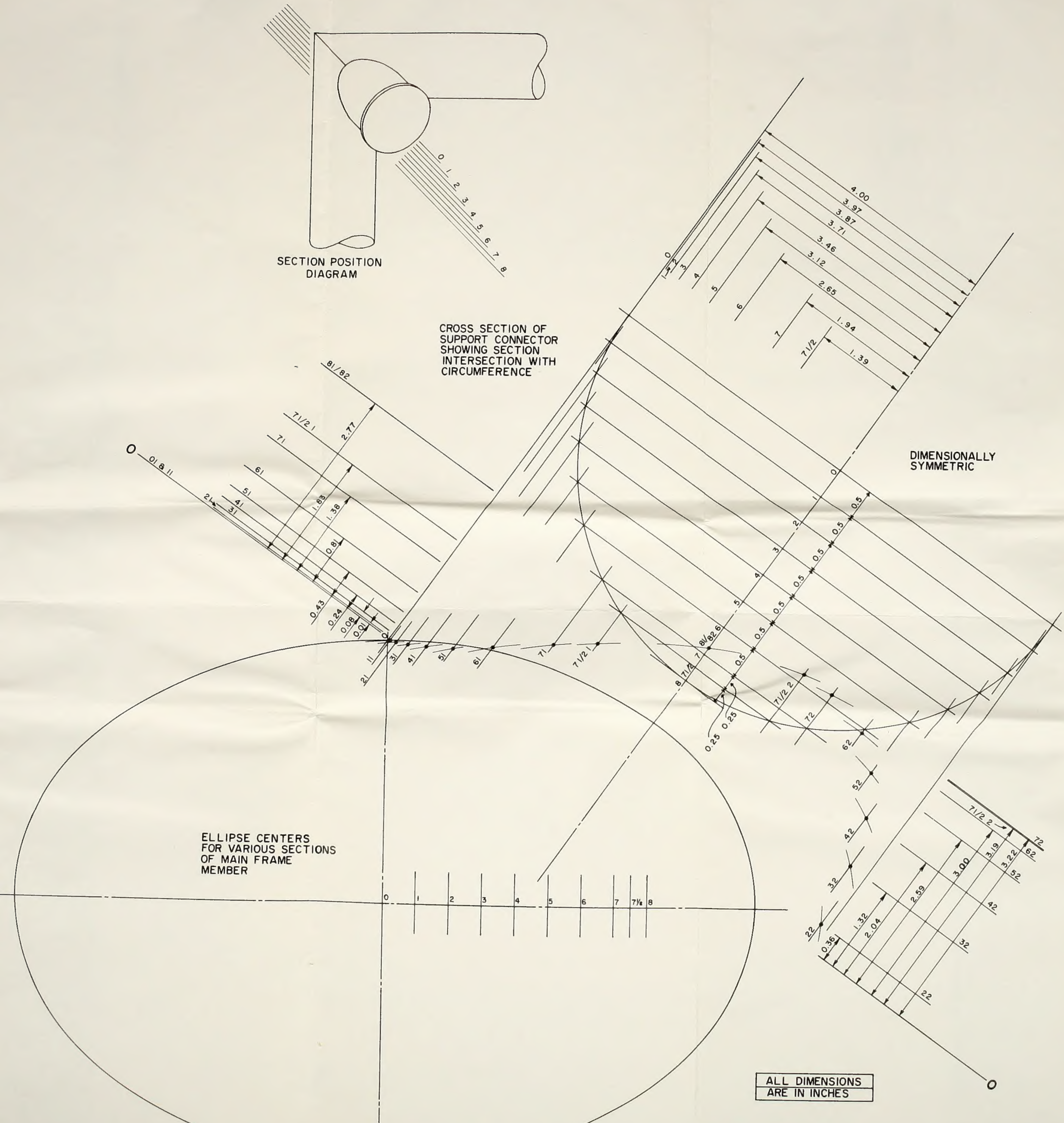

FIG. II -3-5
CONSTRUCTED SECTIONS
FOR CUTTING PATTERN
FOR PIECE II-3-4

Thesis
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c.1
Hooks
152800
Submersible oriented
platform for deep ocean
sediment studies
(SOPDOSS).